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Bureau des programmes solaires

THE SIMAIR COMPUTER PROGRAM
USER'S MANUAL

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NOMENCLATURE

A	total collector area
A_c	area of one collector
A_{fm}	surface area of fibre matrix
A_D	drying area of material
b	spacing between upper and lower absorber plates
C_p	specific heat
C_{p_a}	specific heat of air
C_{p_e}	effective specific heat of air
C_{p_w}	specific heat of water
E_{hx}	effectiveness of heat exchanger
F'	collector efficiency factor $U_o/(U_o + U_L)$
F_R	collector heat removal factor
$F_R\tau\alpha$	collector transmission-absorption coefficient
$F_R U_L$	collector heat loss coefficient
H	measured hourly solar radiation
H_b	beam hourly solar radiation
H_{bT}	beam hourly solar radiation on a tilted surface
H_d	diffuse hourly solar radiation
H_{dT}	diffuse hourly solar radiation on a tilted surface
H_{ex}	extraterrestrial hourly solar radiation
h_{fg_d}	latent heat of vapourization at the dry bulb temperature
h_{fg_w}	latent heat of vapourization at the wet bulb temperature
h_d	mass transfer coefficient
h_{ai}	enthalpy of air
h_{pf}	plate to fluid convective heat transfer coefficient
h_r	radiative heat transfer coefficient
H_r	reflected hourly solar radiation
H_T	solar radiation on the tilted surface
I	total incident solar radiation
k	conductivity of air
K	incidence angle modifier
K_T	clearness index

L	length of drier
L	length of collector in flow direction
$\dot{m}$	mass flow rate
$\dot{m}_a$	mass flow rate of air
$\dot{m}_D$	mass flow rate of dry material
m_D	mass of material
M_D	equilibrium moisture content based on dry weight
M_W	equilibrium moisture content based on wet weight
n	number of air flow passage channels
N	day number of the year
Nu	Nusselt Number
Pr	Prandtl Number
Q_{heat}	energy output from heater
Q_S	solar heating contribution
Qu	useful heat obtained from collector
R	drying rate
R_b	ratio of beam radiation on tilted surface to horizontal surface
Re	Reynolds Number
s	collector slope
S_c	solar constant
T_1, T_2	temperature of upper and lower plates of the air channel
T_a	ambient temperature
Ta_d	dry bulb ambient temperature
Ta_w	wet bulb ambient temperature
T_{ave}	average temperature that collector losses heat from
T_{co}	collector outlet temperature
T_{d_i}	inlet temperature to drier
T_{d_o}	temperature of air leaving drier
T_{fi}	collector fluid inlet temperature
T_{fm}	average collector fluid temperature
T_{hx_o}	temperature out of heat exchanger
T_{plate}	average temperature of the collector absorber plate
U_L	collector heat loss coefficient
U_o	total plate to fluid heat transfer coefficient

W	humidity ratio of air
W_{ai}	humidity ratio for outside air
W_{ao}, W_o	outlet humidity ratio
W_{ai}, W_i	inlet humidity ratio
W_c	critical humidity ratio
W_{s_w}	saturated humidity ratio at the wet bulb temperature
x	matrix constant
x_e	equilibrium moisture content
x_f	desired final moisture content of material
x_i	initial moisture content of material based on dry weight
y	matrix constant

Greek Symbols

α	solar absorptivity
δ	solar declination
γ	azimuth angle
ϕ	latitude
π	3.14159
μ	absolute viscosity
ρ	density
σ	Stefan-Boltzman constant
ω	hour angle
$\tau\alpha$	transmission-absorption product
$(\tau\alpha)_e$	effective transmission-absorption product

1. INTRODUCTION

The SIMAIR computer program was developed by Enermodal Engineering Limited to simulate the performance of several air-based solar heating systems. It is a combination of three previously developed programs: VENTAIR, DRYAIR and ZEROAIR. The SIMAIR program can simulate the performance of air-based solar heating systems which:

- preheat building ventilation or process air,
- dry crops or other materials,
- space heat buildings (with no thermal storage unit).

SIMAIR is the only general purpose computer program capable of simulating the three system types described above. Performance calculations use a one hour time step to ensure maximum accuracy. The program can modify the test data for different flow rates in parallel plate, V-corrugated and fibre matrix air-based collectors.

In solar preheated ventilation or process air systems, outside air is brought through the collectors before going to the conventional air heating and distribution system. If the sun is shining the air will be heated thereby reducing the heating load of the conventional unit. No thermal storage unit is used.

Solar crop or material drying systems are similar to ventilation air systems except that they are used to remove water vapour instead of supplying fresh air. These systems also have no thermal storage unit and use outdoor air as supply to the collectors.

Zero-storage solar space heating systems recirculate building air through air-based solar collectors. This system would only operate when there is a net heat gain at the collectors and the building requires heating. There is no thermal storage unit; excess solar gains are stored in the mass of the building.

The program does, however, have several restrictions. An important restriction on the use of the program for ventilation systems is that the air demand must be constant over the hour. A system not modelled by the SIMAIR program, for example, is a bathroom ventilation system operated by a light switch or short duration timer. For drying systems, the total moisture content of the material in the drier must be relatively constant. This implies a steady movement of material through the drier, since in a batch process the total material moisture content is constantly decreasing. The user must also know the drying characteristics of the material (e.g. equilibrium moisture content and critical moisture content) and of the drier (e.g. moisture removal rate at test conditions). The mechanisms governing the rate of material drying are extremely complex. Although the solar collector energy output can be estimated fairly accurately, the user should expect deviations between the predicted and actual quantities of material dried.

The SIMAIR computer program will provide a reasonable estimate of the thermal performance of air-based solar heating systems provided that the user supplies parameters within the program limitations. All users should read and understand this manual before using the program.

2. SYSTEM DESCRIPTIONS

2.1 Solar Preheated Ventilation or Process Air Systems

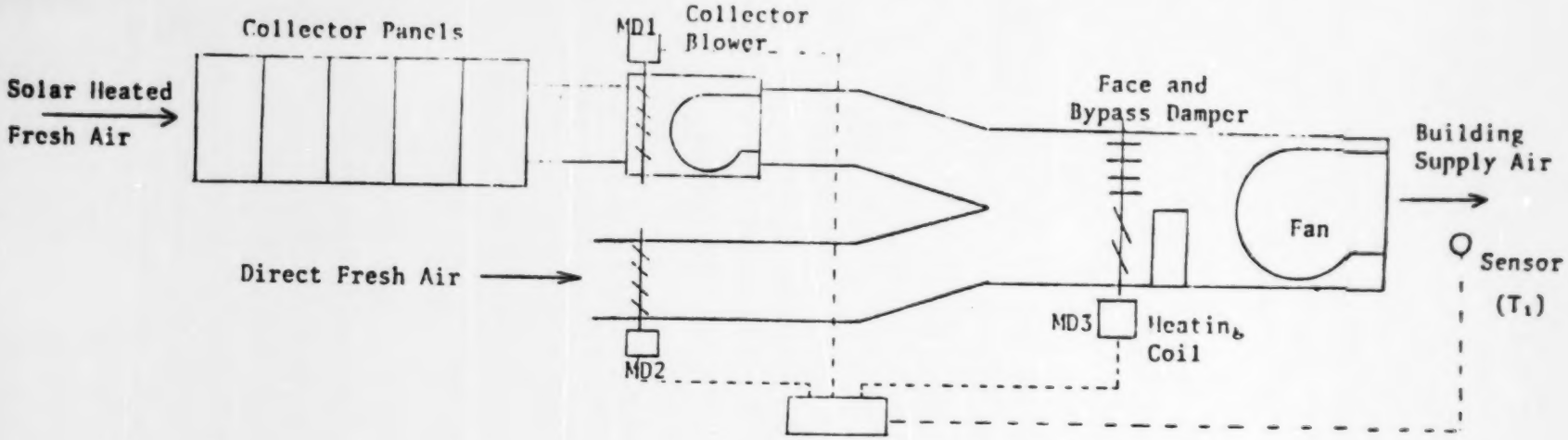
In many buildings a high rate of change of ventilation air is necessary for occupant health or proper equipment operation. Such buildings include hospitals, vehicle maintenance garages, indoor swimming pools and commercial kitchens. The volumes of air involved are often quite large (6 air changes/hour) and correspondingly large quantities of energy are required to raise the air temperature to the building temperature. This air is normally heated by steam coils or electric resistance heaters. Many industrial processes also require large volumes of air for drying and/or heating. Such processes include paint drying, food drying, combustion air preheating and sterilization/drying of bottles and cans.

To reduce energy consumption, heat recovery equipment can be used to transfer some of the heat from the exhaust air to the fresh air. Heat recovery equipment, however, has several limitations. The equipment suffers from frost build-up in sub-freezing conditions, fouling from kitchen grease, and requires modifications if the exhaust air contains toxic chemicals. The installation of this equipment can be expensive in retrofit situations if return air duct work must be added.

Use of air-based solar collectors to preheat ventilation or process air has been proposed as a viable alternative to heat recovery equipment in many applications. Figure 1 shows a system schematic for a typical solar preheated process or ventilation air system.

In a conventional make-up air unit, fresh (outside) air is taken into the building by a roof top fan. A bypass damper automatically adjusts the fraction of the intake air that passes through the heating coil to maintain the required inlet air temperature. Exhaust air is expelled to the outdoors by a separate system. In a solar preheated ventilation system outside air is brought through solar collectors before entering the make-up air unit. If the sun is shining then the air will be heated, thereby reducing the heating load on the heating coil.

Fig.1 System Schematic



T_{bg} is the desired inlet temperature to the building or process

Sequence of Operation							
Stage	Tamb	Damper 1 (MD1)	Damper 2 (MD2)	Collector Blower	MD3	Heating Coil	T_1
1	> T_{bg}	C	O	OFF	bypass	OFF	T_{bg}
2	< T_{bg}	O	O	ON	bypass	OFF	T_{bg}
3	< Tbg	O	C	ON	bypass and coil	ON	T_{bg}

Solar preheated ventilation air systems differ from conventional solar space heating systems in two important areas. First, the inlet air to the collector is the outside or ambient air, as opposed to the building air. This means that the collector will operate at a lower temperature and a higher efficiency. Second, because the solar energy is delivered immediately to the heating load there is no need for thermal storage. If the solar heating contribution of the total heating load is below 35%, this system can deliver more energy per square metre than a conventional solar space heating system (see reference 7). If a solar contribution significantly higher than 35% is required, thermal storage should be considered, although the increased cost may not be justified. A side effect of eliminating the storage unit is the simplification of system control and ductwork.

Solar preheated ventilation air systems have three modes of operation (excluding complete shut down). These modes are summarized in Figure 1 and described below.

(1) No Solar Heating

If the ambient air temperature is greater than the required building or process air temperature, no heating of the air is required. In order to eliminate an unnecessarily high cooling load, all air enters the direct fresh air intake bypassing the solar collectors. The air bypasses the heating coil and is handled by conventional air conditioning equipment before entering the space.

(2) All Solar Heating

If the ambient air temperature drops below the building or process temperature, some air would flow through the collectors. This is accomplished by partially opening damper 1 and partially closing damper 2 (see Figure 1). Flow would continue to be diverted until the supply air temperature reaches the building or process temperature. In this operation all the heat necessary to bring the ambient air temperature up to the building or process temperature is supplied by the solar collectors.

(3) Partial Solar Heating

As the ambient temperature continues to decrease, more air is taken through the collectors and less through the direct fresh air intake. A point will be reached when the collectors are operating at maximum air flow (damper 1 fully open, damper 2 fully closed) and the building supply air temperature (T1) is less than the required temperature. In this case, additional heating is supplied by the heating coil.

It should be noted that solar collectors have a maximum flow rate where the incremental solar heat gain is less than the incremental fan energy consumption. Thus in many designs, with the maximum flow rate of air flowing through the collectors, some air would be supplied through the direct fresh air intake in order to meet the required ventilation rate.

In Mode 2 all the available solar heat has not been used. Theoretically, this heat could be used to heat the building (i.e. by supplying ventilation air at a temperature higher than the building temperature). However, this is a much more difficult control problem requiring zone heat control for ventilated and unventilated areas. Control feedback to the solar ventilation system would be necessary in the ventilated areas to prevent overheating. Such a control strategy is not considered applicable for space heating.

2.2 Solar Crop/Material Drying Systems

Drying of materials is necessary in many industrial and agricultural processes. To prevent spoilage many crops must be dried including corn, wheat, sugar cane, etc. Many foodstuffs are dried for long term storage or for transportation including milk, flour, eggs, etc. Drying is also a critical step in many industrial processes. Materials that require drying include leather, fibres, paper and wood.

The rate of drying is dependent on the drier design. Drier designs can be divided into two general types: continuous and batch. In batch driers, the drier is filled with wet material and not emptied until all of the material

reaches the desired minimum moisture content. In continuous driers, as material reaches the desired minimum moisture content it is removed and wet material added. Material need not be removed continuously but often enough that the total moisture content of the material in the drier stays relatively constant.

In batch driers the drying rate is constantly changing because the average moisture content of the material in the drier is changing. The rate of change in the drying rates is so variable that it is almost impossible to predict (1). For this reason batch driers are not handled by the SIMAIR program.

Continuous driers can be subdivided into three types: counter-flow, cross-flow and parallel-flow (similar to heat exchanger types). In counter-flow driers, air and material flow in parallel but opposite directions. In parallel-flow driers, air and material flow in the same direction. Air and material flow in perpendicular directions in a cross-flow drier. Like heat exchangers, a counter-flow arrangement is the most efficient and parallel-flow the least efficient. Because of the inefficiency of parallel-flow driers they are not handled by the SIMAIR program.

Besides the drier design, the drying rate is dependent on the relative humidity of the air. Relative humidity is the ratio of the weight of water in the air to the maximum weight of water that the air can hold (saturated conditions). Relative humidity and drying rate are inversely related. Thus, the drying rate decreases as the relative humidity increases until the relative humidity of air is equal to the relative saturation of the material. At this condition, referred to as the equilibrium moisture content of the material, no moisture is transferred. The value of the equilibrium moisture content depends on the material, the relative humidity of the air and to a lesser extent temperature.

The most common method of decreasing the relative humidity (increasing the drying rate) is to heat the air. The more the air is heated, the greater the drying rate is. For most crops and materials, however, there is an upper limit on the inlet air temperature above which the crop or material may crack.

A typical air-based solar crop/material drying system is shown in Figure 2. The system consists of air-based solar collectors, fan, auxiliary heat and material drier. The operation of this system is quite simple. When the drier is operating, outside air is brought in the air inlet. If a heat exchanger is placed before the collector, heat from the drier exhaust is transferred to the outdoor air. The air then passes through the collectors (even if the sun is not shining) to be heated by solar energy. If the air leaving the collector is too hot, then the air flow through the collector is reduced and outside air is mixed with the collector outlet air. If the heat exchanger is placed after the collector, heat from the drier exhaust is transferred to the air. The air then passes through the heater. There are two types of heater: constant temperature and constant energy input. For the constant temperature heater the outlet temperature is controlled at a constant value. Conversely, the constant energy input heater would have a variable temperature, fluctuating with the collector outlet temperature.

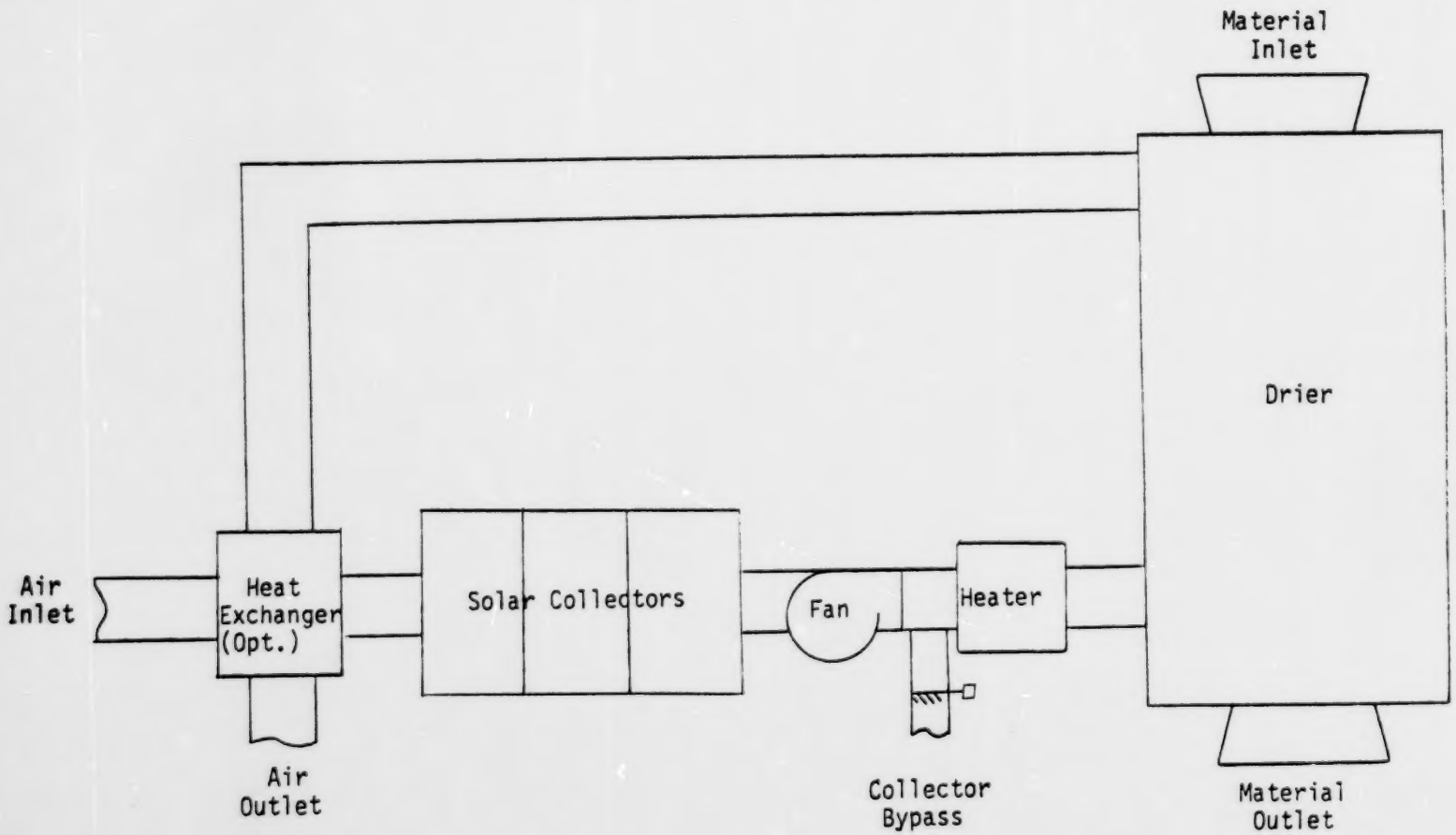
The hot dry air passes through the drier, picking up moisture from the material. As the material is dried to the desired condition it is removed and wet material added.

2.3 Zero Storage Solar Space Heating Systems

Eliminating the storage unit from the solar heating system significantly reduces the cost and complexity of the system. When solar heat is available, it is blown directly into the building. The thermal mass of the building can store some solar heat, however, on sunny days some solar heat may have to go uncollected to prevent the building from overheating.

In general, this type of system is best suited to buildings that can be allowed to fluctuate in temperature (e.g. warehouses). Buildings that require precise temperature control and have high internal heat gains during the day are not suited to this type of solar heating system. Because there is no storage of heat (other than in the building structure and its contents), the solar heating system can supply very little of the nighttime heating load. As such the system should be sized to meet under 40% of the heating load. A large system would be inefficient and therefore expensive.

Figure 2: Typical Solar Material Drying System (Counter-Flow)



A zero storage air-based solar space heating system is shown in Figure 3. The system consists of air-based solar collectors, supply and return ductwork, fan and control system. In many cases the collectors can be mounted directly on the south wall of a building with no need for collector support racks or supply and return ductwork. If collector air is blown directly into the building it may be necessary to install a ceiling fan to mix the building air.

The control system for this type of system is quite simple. There are only two modes of operation: system on and system off. If the temperature of the collector is greater than that of the building, air is circulated through the collectors. The fan would continue to operate until the collector temperature drops below the building temperature or the maximum allowable building temperature is exceeded. On spring and fall days when the solar gain exceeds the heating load, the system will constantly cycle on and off so as not to exceed the maximum building temperature.

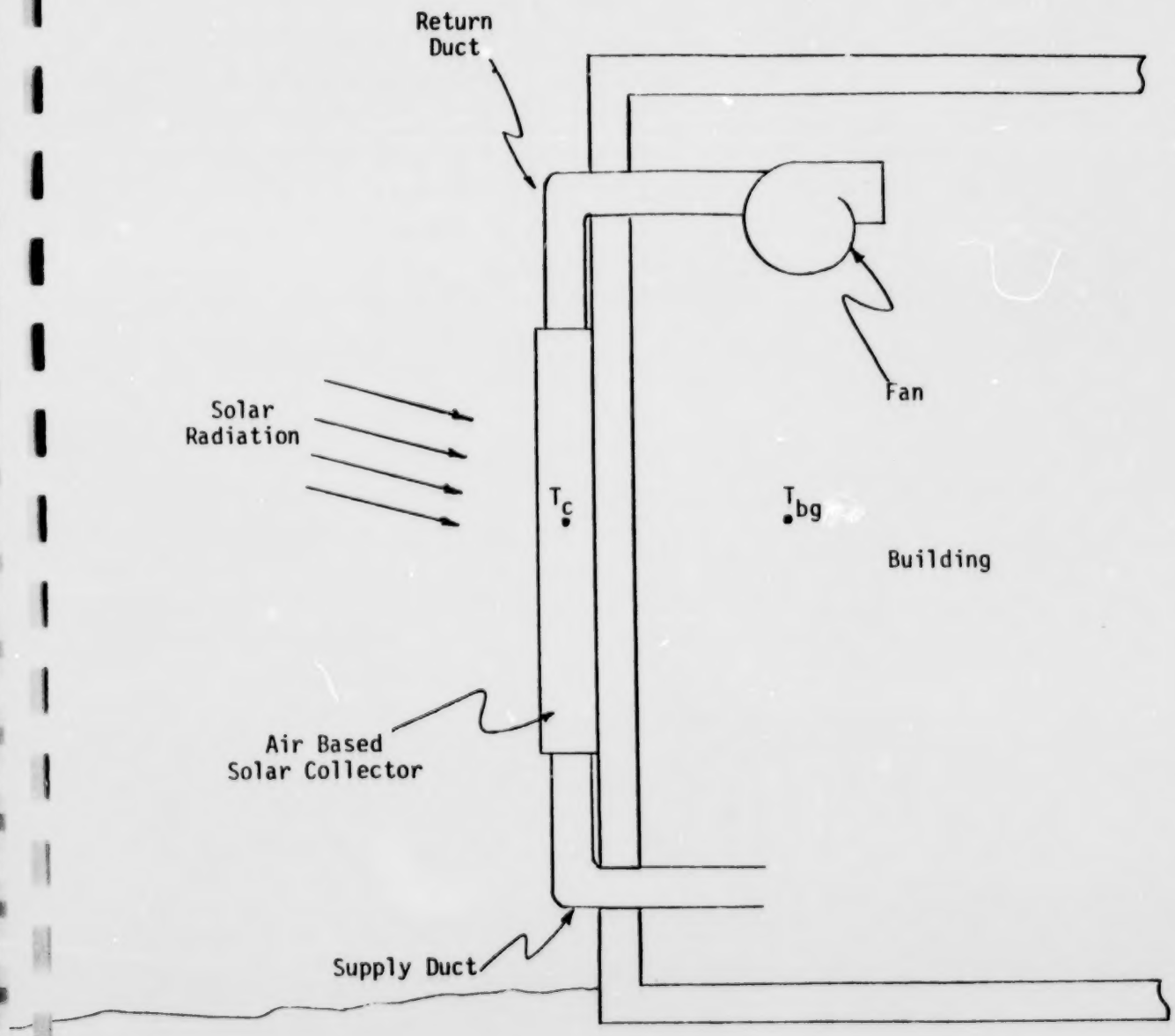


Figure 3: Zero Storage Air-Based Solar Space Heating System

2.4 Using the SIMAIR Program

The SIMAIR computer program is relatively easy to use. The program has been written so that the user enters the system description interactively with the computer. The program makes some fundamental checks on the suitability of the input parameters. If a parameter appears unrealistic, a warning is printed although the program accepts the value.

Before the program can be run hourly weather data (solar radiation on a horizontal surface, dry bulb and wet bulb temperatures) must be available. This data can be obtained from Atmospheric Environment Service. The first step in the program execution is to calculate solar radiation on the collector surface and store it in a scratch file. During the simulation the program accesses the scratch file. If collector slope and orientation remain the same on successive runs, the same scratch file is used. If the values are changed, the scratch file is automatically recalculated.

When the user accesses the program, a series of questions concerning the program will have to be answered. These questions are shown in Section 2.3. When the questions have been answered, the user has the choice of several commands:

C or CHANGE (or CL)
D or DEFAULT
L or LIST
M or MODIFY
R or RUN
S or STOP
T or TITLE

The uses of these commands is discussed below.

CHANGE this command is used to change the program input data. A list of the input data and their meanings is given in Section 3.1. The command is of the form C or CHANGE Parameter # + New Value
e.g. C C3 = 50
Several changes can be made on one line with our without the equal sign
e.g. C C4 = 0.0, T1 3

Parameter changes can be listed at the same time changes are made by the CL command

e.g. CL C1 10

DEFAULT this command is used to re-initialize all the input parameters to their default values. There are no options with this parameter.

LIST this command is used to list either all the input parameters, all the parameters of one section or an individual parameter.

L or LIST ALL - lists all parameters

L C - lists all collector parameters

L C3 - lists collector slope

MODIFY this command is used to modify the system type and the default collector characteristics i.e. to change the collector type.

RUN this command is used to run the program. If the option "P" is used a copy of the output will be sent to the printer. If the option "D" is specified a detailed hour-by-hour output is produced. If the option "Q" or "Quick" is specified the input parameter listing is not given. If a simulation period other than one year is required, the first day of the simulation followed by the length of the simulation should be entered

e.g. RUN or R P 31, 28 - this would execute the program starting January 31st and run for 28 days. A copy of the output would be sent to the printer.

STOP this command stops the program.

TITLE this command is used to insert a title into the program output. The same title will be printed on every successive run unless it is changed.

e.g. T or TITLE THIS IS RUN NUMBER 1.

```
**************************************
*                                    *
*              SIMAIR                *
*     VERSION 1.0  JANUARY,1984      *
*                                    *
**************************************


DO YOU WISH INSTRUCTIONS? (Y/N) :n
 IS PROCESSED WEATHER DATA AVAILABLE (Y/N) ? y
 ENTER SYSTEM TYPE :
    1 - VENTILATION OR PROCESS AIR HEATING
    2 - CROP OR MATERIAL DRYING
    3 - ZERO STORAGE SPACE HEATING ? 1
 ENTER SOLAR COLLECTOR TEST DATA :
    1 - AMHERST 200 : JAN,1982
    2 - CONSERVAL : APRIL,1983
    3 - LPC : MAY,1982
    4 - WATERSHED A100 : NOV,1982
    USE ZERO TO KEEP EXISTING DATA  ? 2

C,D,L,M,R,S,T,?: c c1 100.
THE NUMBER OF CHANGES MADE =    1

C,D,L,M,R,S,T,?: cl c6 35.
    C6  COLLECTOR FLOW RATE PER UNIT AREA (L/(SEC*M2)).    35.00
THE NUMBER OF CHANGES MADE =    1

C,D,L,M,R,S,T,?: t sample run #1

C,D,L,M,R,S,T,?: r 1,90
```

**** SIMAIR INPUT DATA ****

SAMPLE RUN #1

SIMULATION PERIOD - FIRST DAY : 1 LAST DAY : 90

C. COLLECTOR DATA

C1	NUMBER OF COLLECTORS	100.00
C2	NUMBER OF COLLECTORS IN SERIES	1.00
C3	COLLECTOR FLOW CHANNEL HEIGHT (MM)	108.00
C4	COLLECTOR SLOPE (DEGREES)	45.00
C5	COLLECTOR ORIENTATION (SOUTH=0,EAST=+, DEG.)	.00
C6	COLLECTOR FLOW RATE PER UNIT AREA (L/(SEC*M2))	35.00
C7	PERCENT COLLECTOR LOOP AIR LEAKAGE	.00
C8	FAN POWER CONSUMPTION PER UNIT AREA (W/M2)	10.00

T. COLLECTOR TEST DATA

T1	TYPE: 1-PL.PLATE, 2-FIBRE MATRIX, 3-V-CORRUGATED	3.00
T2	FLOW: 1-BELOW ABSORBER, 2-ABOVE ABSORBER, 3-BOTH	2.00
T3	GROSS AREA OF ONE COLLECTOR (M2)	1.30
T4	NUMBER OF COLLECTORS IN SERIES WHEN TESTED	1.00
T5	FR-TAU-ALPHA (TEST)	.59
T6	FR-UL (W/(M2*C)) (TEST)	10.61
T7	COLL. TEST FLOW RATE PER UNIT AREA (L/(SEC*M2))	30.00
T8	TRANSMISSION-ABSORPTION PRODUCT	.90
T9	RATIO OF APERTURE TO GROSS AREA	.97
T10	COLLECTOR FLOW CHANNEL HEIGHT WHEN TESTED (MM)	108.00
T11	RATIO OF ABORBER AREA TO COLLECTOR AREA	2.00
T12	COLLECTOR FLOW CHANNEL LENGTH PER COLLECTOR (M)	2.44

T13 INCIDENT ANGLE MODIFIER

I :	1	2	3	4	5
THETA :	15.000	30.000	45.000	60.000	75.000
ALT :	.000	1.000	1.020	1.040	.940

V. VENTILATION LOAD DATA

V1	BUILDING OR PROCESS TEMPERATURE (C)	20.00
V2	DAYS PER WEEK AT AIR FLOW PROFILE 1	7.00
V3	DAILY AIR FLOW RATE AT PROFILE 1 (K-M3/DAY)	100.00
V4	DAILY AIR FLOW RATE AT PROFILE 2 (K-M3/DAY)	100.00
V5	HOURLY AIR FLOW RATE PROFILE 1 (%)	100.00

4.1600	4.1600	4.1600	4.1600	4.1600	4.1600
4.1700	4.1700	4.1700	4.1700	4.1700	4.1700
4.1700	4.1700	4.1700	4.1700	4.1700	4.1700
4.1700	4.1700	4.1700	4.1700	4.1700	4.1700

V6	HOURLY AIR FLOW RATE PROFILE 2 (%)	100.00

2.2500	.0000	.0000	.0000	.0000	.0000
1.5000	4.6500	7.2500	8.4900	6.9000	4.5000
3.6000	5.1000	2.7000	2.4000	2.1000	3.7500
6.7500	11.6000	9.6000	6.9000	5.4600	4.6500

CITY DATA

LOCATION .....OTTAWA

LATITUDE OF LOCATION (DEG.)..... 45.5

MODIFIED COLLECTOR CHARACTERISTICS

FRTA - ADJUSTED = .62
FRUL - ADJUSTED =11.27 W/(M2-C)
REQUIRED CORRECTION FACTOR = 1.05

THE MAXIMUM FLOW RATE TO COLLECTOR AREA IS 35.0 LITRES/(M2*S)

****
PERFORMANCE PREDICTION
****

MON	SOLAR AVAIL. (GJ)	SOLAR COLLECT (GJ)	AUX. ENERGY (GJ)	HEATING LOAD (GJ)	FAN POWER (GJ)
JAN	49.501	18.341	104.367	122.704	1.282
FEB	42.674	15.118	78.481	93.598	1.310
MAR	81.215	24.963	64.294	89.256	1.713
YR	173.39	58.42	247.14	305.56	4.31

ENERGY SAVINGS DUE TO SOLAR (GJ) : 54.12

ENERGY SAVINGS DUE TO SOLAR (%) : 17.71

AVERAGE SOLAR SYSTEM COLLECTION EFFICIENCY (%) : 31.21

SOLAR CONTRIBUTION PER SQUARE METRE (GJ/M2) : .416

C,D,L,M,R,S,T,?: s

3. SYSTEM INPUT PARAMETERS

The following sections describe the parameters used in the SIMAIR computer program. The default value for each parameter is included with the definition.

3.1 Definition of Input Parameters

There are seven separate sections of input data:

C - collector data
T - collector test data
M - material drying characteristics
L - drying load data
D - drier characteristics
B - building heat load data
V - ventilation or process air data

The sections required for the three system types are shown below:

Section	Ventilation or Process Air Heating	Crop or Material Drying	Zero Storage Space Heating
C	X	X	X
T	X	X	X
M		X	
L		X	
D		X	
B			X
V	X		

The parameters for each of these sections are discussed below.

Collector Data

C1 NUMBER OF COLLECTORS (2)

The number of collectors in the solar heating system.

C2 NUMBER OF COLLECTORS IN SERIES (1)

The number of collectors that are connected in series as opposed to in parallel.

C3 COLLECTOR FLOW CHANNEL HEIGHT (MM) (6)

The average height of the flow channel to be used in the installation in millimetres, for V corrugated absorbers it would be one half the depth of the corrugations plus the height of the absorber above the rear absorber (for T2=2) or below the glazing (for T2=1). In most cases this height will be equal to parameter T10.

C4 COLLECTOR SLOPE (DEGREES) (45)

The angle that the collector is tilted from the horizontal in degrees. This parameter must be between 0 and 90. If this parameter is different from the value on the scratch weather file, the weather data will automatically be reprocessed.

C5 COLLECTOR ORIENTATION (SOUTH=0 EAST=+, DEG) (0)

The number of degrees that the collector is oriented off due south (east is positive, west is negative). This parameter must be between -90 and 90. If this parameter is different from the value on the scratch weather file, then the weather data will automatically be reprocessed.

C6 COLLECTOR FLOW RATE PER UNIT AREA (L/(SEC*M2) (10)

The air flow rate per unit collector area through the collectors for the proposed system in litres per sec per square metre of collector. Note: This is also the flow rate through the drier.

C7 PERCENT COLLECTOR LOOP AIR LEAKAGE (5)

Percent of the collector flow rate that leaks into the collector loop (i.e. collector under negative pressure). For a collector loop under positive pressure (air leaks out) use a negative percentage. A well designed and installed system should have a leakage rate of under 10%.

C8 FAN POWER CONSUMPTION PER UNIT AREA (W/M2) (10)

The incremental energy required to operate the collector fan in watts per square metre of collector. This should not include the fan power that would be required to circulate the air if there was no solar heating system.

Collector Test Data

T1 TYPE: 1 - PARALLEL PLATE, 2 - FIBRE MATRIX, 3 - V-CORRUGATED (1)

The type of collector used in the system where 1 is a parallel plate, 2 is a fibre matrix and 3 is a V corrugated collector.

T2 FLOW TYPE: 1 - BELOW ABSORBER, 2 - ABOVE ABSORBER, 3 - BOTH (1)

Location of the air flow passage where 1 is between the top and rear absorber plates, 2 is between the glazing and absorber plate and 3 is both of these passages.

T3 GROSS AREA OF ONE COLLECTOR (M2) (2)

The gross or outside area of one collector in square metres.

T4 NUMBER OF COLLECTORS IN SERIES WHEN TESTED (1)

Number of collectors connected in series when it was tested, typically one.

T5 FR-TAU-ALPHA (TEST) (0.5)

The $F_R(\tau\alpha)_e$ of the collector when tested by a certified laboratory (based on gross collector area).

T6 FR-UL (W/M2*C) (TEST) (4.0)

The $F_R U_L$ of the collector when tested by a certified laboratory (based on gross collector area) in $W/(m^2\,°C)$.

T7 COLL. TEST FLOW RATE PER UNIT AREA (L/SEC*M2) (10.)

The air flow rate through the collector at test conditions in litres per second per square metre of collector.

T8 TRANSMISSION-ABSORPTION PRODUCT (0.9)

The $(\tau\alpha)$ effective of the collector. This can be estimated as 1.01 times $\tau\alpha$ for a single glazed collector where τ is the glazing solar transmission and α is absorber solar absorptivity.

T9 RATIO OF COLLECTOR APERTURE TO GROSS AREA (0.9)

The ratio of the collector aperture area (i.e. window area) to the gross area. Typically this value is usually close to 0.9, actual values can be obtained from collector data sheets.

T10 COLLECTOR FLOW CHANNEL HEIGHT (TEST) (MM) (6)

The average height of the collector flow channel when tested (see C3 for a complete description). In general decreasing the flow channel height increases the collector F_R although this will result in a higher collector pressure drop.

T11 RATIO OF ABSORBER AREA TO COLLECTOR AREA (1)

Surface area of absorber divided by area of collector. This parameter is normally one for flat plate and fibre matrix collectors. For V-corrugated collectors this parameter is equal to $\sqrt{1 + (2dn/W)^2}$ where d is the depth of the corrugations, n is the number of corrugations and W is the width of the flow path (i.e. width of the collector).

T12 COLLECTOR FLOW CHANNEL LENGTH PER COLLECTOR (M) (1)

The distance or length that the air stream is in contact with the absorber plate for one collector. In most collectors this will be the length of collector in the flow direction. In some cases, however, collectors are designed so that the collector length is much shorter than the collector such as in the overlapped glass plate collector.

T13 INCIDENT ANGLE MODIFIER

The user must specify 5 incident angle modifiers for 5 different solar incident angles. Unless specified by the user, the program assumes that the incident angle modifier is 0 for an angle of 90°.

Material Drying Characteristics

Typical values for these parameters are given in Section 3.3.

M1 % EQUIL MOISTURE CONTENT AT 10 RH AIR (DRY WT.) (3.)

Percentage of material that is water when at equilibrium with air at 10% relative humidity based on dry weight (see Section 3.3).

M2 % EQUIL MOISTURE CONTENT AT 70 RH AIR (DRY WT.) (10.)

Percentage of material that is water when at equilibium with air at 70% relative humidity based on dry weight (see Section 3.3).

M3 % CRITICAL MOISTURE CONTENT (DRY WT.) (40.)

Percentage moisture content at which the drying rate begins to decrease based on dry weight (see Figure 4). If no information is available a conservative estimate would be the equilibrium moisture content at 100% relative humidity air (see Section 3.3).

M4 SPECIFIC HEAT OF DRY MATERIAL (KJ/(KG*C)) (2.)

Specific heat of the dry material, i.e. no water content.

Drying Load Data

L1 MASS OF MATERIAL TO BE DRIED (KG/DAY) (0 = MAX)

Mass of material based on dry weight to be dried per day. The system will operate as long as required to dry this quantity. If zero is entered, the amount of material that can be dried in the time specified is calculated.

L2 % INITIAL MOISTURE CONTENT (KG-WATER/KG-DRY MAT.) (70.)

Initial percentage of material that is water based on dry weight.

L3 INLET TEMPERATURE OF MATERIAL (C) (20.)

The average temperature of the material when it is added to the drier.

L4 % FINAL MOISTURE CONTENT (KG-WATER/KG-DRY MAT.) (20.)

Final or desired percentage of material that is water based on dry weight.

L5 NUMBER OF DAYS OF OPERATION PER WEEK (7.)

Number of days per week that the drier operates.

L6 HOURS OF OPERATION PER DAY, 0 = REQUIRED TIME (8.)

The number of hours per day that the drier is to operate. If zero is entered, the number of hours required to dry the amount of material specified in L1 is calculated.

L7 STARTING HOUR FOR DRYING OPERATION (8.)

The first hour that the drier is to operate, where 0 = midnight, 8 = 8:00 A.M., 16 = 4:00 P.M.

Drier Characteristics

D1 DRIER DESIGN: 1 - COUNTER FLOW, 2 - CROSS FLOW (1)

The design of the drier.

D2 DRIER TYPE: 1 - CONSTANT ENERGY INPUT, 2 - CONSTANT TEMP. (2)

The type of heater used in the drier: either constant drier inlet temperature (electric) or constant heat (gas or oil).

D3 MAXIMUM AIR INLET TEMPERATURE (C) (30)

The desired drier inlet temperature if D2 is 2, or the maximum allowable temperature before the heater will shut off if D2 = 1.

D4 IF D2 = 1, AUX. HEATER OUTPUT (KW) (10)

The output from the auxiliary heater. If D2 = 2 this parameter is not used.

D5 DRIER H.X. LOCATION 0 - NONE, 1 - BEFORE, 2 - AFTER COLLECTOR (0)

Location of drier heat exchanger where 0 indicates no heat exchanger is used, 1 - the heat exchanger is before the collectors, 2 - the heat exchanger is after the collectors. Note: it is possible to have a negative solar contribution if the heat exchanger is before the collectors since heat recovered by the heat exchanger could be lost by the collector.

D6 EFFECTIVENESS OF DRIER HEAT EXCHANGER (0.)

The effectiveness or efficiency of the heat exchanger used to reclaim heat from the drier exhaust to preheat outside air entering the drier. This parameter is normally less than 0.8.

D7 - D10 These parameters describe the performance of the drier. These parameters must be for the material and moisture contents given in Section M and L2, L3 and L4.

D7 DRIER TEST FLOW RATE (M3/S) (1)

The flow rate that the drier was tested at. It need not be the same as the proposed system flow rate (see parameter C5).

D8 DRIER TEST INLET TEMPERATURE (C) (20)

The inlet temperature to the drier when it was tested (i.e. the temperature after the heater as shown in Figure).

D9 TEST INLET RELATIVE HUMIDITY (%) (20)

The relative humidity of the air at the drier inlet.

D10 TEST OUTLET RELATIVE HUMIDITY (%) (70)

The relative humidity of the air at the outlet from the drier (before the heat exchanger).

Heating Load Data

B1 BUILDING HEAT LOSS COEFFICIENT (W/C) (300.)

The heat loss coefficient or "UA" of the building. Section 3.4 gives a method of estimating this parameter.

B2 MINIMUM DAYTIME BUILDING TEMPERATURE (C) (20.)

The minimum allowable temperature of the building during the day (i.e. the furnace thermostat set point).

B3 MINIMUM NIGHTTIME BUILDING TEMPERATURE (C) (16.)

The minimum allowable temperature of the building during the night (i.e. the furnace thermostat set point at night).

B4 MAXIMUM ALLOWABLE BUILDING TEMPERATURE (C) (25)

The maximum desired building temperature. That is the temperature above which the building's occupants would be uncomfortable.

B5 BUILDING THERMAL CAPACITANCE (MJ/C) (15.)

The thermal or heat capacitance of the building. Most buildings have a thermal capacitance of 0.1 MJ/C for every square metre of floor area, although full warehouses would be slightly higher.

B6 DAILY INTERNAL HEAT GAIN (MJ/DAY) (30)

The average daily internal heat gains from lights, people and solar gains through windows in MJ.

B7 HOURLY INTERNAL HEAT GAIN PROFILE (PERCENT)

24 hourly values of the percent of the daily internal heat gain occurring within that hour. The first value is for the hour 12 midnight to 1 a.m.

Ventilation Load Data

V1 BUILDING OR PROCESS TEMPERATURE (°C) (20.)

The desired air temperature for the building or process (i.e. the temperature that the air must be heated up to).

V2 DAYS PER WEEK AT AIR FLOW PROFILE 1 (7)

Number of days in the week that the air flow rate is at profile 1 (see parameters V3 and V5). The remainder of the week is assumed to be at profile 2.

V3 DAILY AIR FLOW RATE AT PROFILE 1 (K-M3/DAY) (10.)

The total volume of air used each day at profile 1 in thousands of cubic metres.

V4 DAILY AIR FLOW RATE AT PROFILE 2 (K-M3/DAY) (10.)

The total volume of air used each day at profile 2 in thousands of cubic metres.

V5 HOURLY AIR FLOW RATE PROFILE 1 (%)

24 values of the percentage of total daily flow rate at profile 1 that occurs in each hour. The 24 values must add up to 100%.

V6 HOURLY AIR FLOW RATE PROFILE 2 (%)

24 values of the percentage of total daily flow rate at profile 2 that occurs in each hour. The 24 values must add up to 100%.

3.2 Weather Data

At present there is weather data for 46 cities that can be used by the program. These cities are tabulated below. The solar radiation data as supplied by Atmospheric Environment Service is of two types: derived or measured. Measured data is as recorded by their monitoring equipment (with missing data estimated from the previous day's values). Derived data is predicted by using other meteorological data such as rainfall, cloud cover, etc.

City	Province	Latitude (Deg.)	Year	Solar Rad. Derived/Measured
Victoria	B.C.	48.7	1971	D
Prince George	B.C.	53.9	1974	M
Vancouver	B.C.	49.2	1971	M
Summerland	B.C.	49.6	1971	D
Frobisher Bay	N.W.T.	36.8	1975	D
Resolute	N.W.T.	74.7	1971	M
Edmonton	Alta.	53.6	1971	M
Medicine Hat	Alta.	50.5	1971	D
Uranium City	Sask.	59.6	1971	D
Swift Current	Sask.	50.3	1971	D
Saskatoon	Sask.	52.2	1971	D
Churchill	Man.	58.8	1975	D
Brandon	Man.	49.9	1971	D
Winnipeg	Man.	49.9	1971	M
The Pas	Man.	53.8	1971	M

City	Province	Latitude (Deg.)	Year	Solar Rad. Derived/Measured
Thunder Bay	Ont.	48.4	1971	D
Sault Ste. Marie	Ont.	46.5	1971	D
Sudbury	Ont.	46.5	1971	D
Kapuskasing	Ont.	49.4	1966	M
Kingston	Ont.	44.2	1971	D
Muskoka	Ont.	45.0	1971	D
Windsor	Ont.	42.3	1971	D
London	Ont.	43.0	1971	D
Toronto	Ont.	43.7	1971	M
Ottawa	Ont.	45.4	1971	M
Montreal	Que.	45.4	1971	M
Sept Iles	Que.	50.2	1974	M
Quebec	Que.	46.8	1971	D
Sherbrooke	Que.	45.4	1971	D
Riviere du Loop	Que.	47.8	1971	D
Bagotville	Que.	48.3	1971	D
Val D'Or	Que.	84.0	1971	D
Fredericton	N.B.	45.9	1971	M
Charlo	N.B.	48.0	1971	D
Chatham	N.B.	47.0	1971	D
Moncton	N.B.	46.1	1971	D
Saint John	N.B.	45.3	1971	D
Charlottetown	P.E.I.	46.3	1971	D
Truro	N.S.	45.4	1971	D
Halifax	N.S.	44.7	1971	M
Sydney	N.S.	46.2	1971	D
Yarmouth	N.S.	42.8	1971	D
St. John's	Nfld.	47.6	1971	M
Gander	Nfld.	49.0	1971	D
Stephenville	Nfld.	48.5	1971	D
Goose	Nfld.	53.3	1971	M

3.3 Drying Properties of Materials

Table 3.1 contains typical drying properties of several crops and foodstuffs. If the user has drying information on the material being processed, this should be used instead of the data in the table. This data is extracted from reference 2 and the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE) Handbook of Fundamentals (1977) Chapter 10 and is for a material temperature of approximately 25°C. The effect of temperature on these properties is shown in the Appendix for several grains.

3.4 Estimation of Building Heat Loss Coefficient

For new homes the building heat loss coefficient can be estimated by adding up the heat loss from each building wall, window and door and including an estimate of air infiltration. The ASHRAE Handbook of Fundamentals and many heating textbooks show how this can be done. This calculation is normally done for every house in order to size the furnace.

If heating bills are available it is easy to estimate the building heat loss coefficient (UA_b) using the formula

$$UA_b = \frac{11600\ FC\ \eta_f}{DD\ HV} \quad \text{(in W/°C)}$$

where FC is the fuel consumption in litres of oil, cubic metres of gas of kWhr of electricity,

η_f is the seasonal furnace efficiency, typically 0.6 for oil and natural gas and 1.0 for electricity,

HV is the heating value of fuel

oil = 25 l/GJ

natural gas = 27 m^3/GJ

electricity = 278 kWhr/GJ

DD is the yearly total of degree-days below 18°C in °C-days

Vancouver - 3000

Toronto - 4000

Ottawa - 4600

Fredericton - 4600

Winnipeg - 5900

Yellowknife - 8600

Table 3.1: Drying Characteristics of Materials (at 25°C)

	Equilibrium Moisture Content (%)		
	with 10% R.H.	with 70% R.H. based on dry weight[1]	with 100% R.H.
Crops			
Corn, yellow dent	4.6	16.4	28.0
Flaxseed	5.3	17.6	25.0
Sorghum	5.8	16.6	21.0
Soybean	3.3	16.4	20.0
Wheat	6.9	15.5	23.0
Foodstuffs			
White bread	0.5	11.1	23.5
Flour	2.6	12.4	22.8
Starch	2.2	9.2	14.8
Gelatin	0.7	7.6	13.5
Materials			
Cotton, American cloth	2.6	8.1	18.6
Cotton, Absorbent	4.8	22.8	27.3
Wool, Australian	4.7	17.2	26.9
Silk, raw	3.2	11.9	22.3
Linen, dry spun yarn	3.6	9.8	16.4
Hemp, manila and sisal	2.7	11.6	17.8
Rayon, Vicose nitro-cellulose	4.0	12.4	18.2
Rayon, Cellulose Acetate	0.8	3.6	6.3
Paper, M.F. newsprint	2.1	7.2	12.5
Paper, white bond	2.4	8.8	15.6
Paper, kraft wrapping	3.2	10.5	17.2
Leather, sole oak-tanned	5.0	20.6	34.4
Rubber, solid tires	0.11	0.76	1.1
Wood, timber	3.0	14.0	26.5
Soap, white	1.9	16.1	27.8
Tobacco, cigarette	5.4	25.0	66.5
Asbestos fiber, finely divided	0.16	0.62	0.95
Activated charcoal, steam activated	7.1	30.0	34.3
Sulfuric acid, H_2SO_4	33.0	67.0	91.5

[1] The percentage equilibrium moisture content based on dry weight can be calculated from the equilibrium moisture content based on wet weight.

$$M_D = 100\ (M_w)/(100 - M_w)$$

Typical values for the building heat loss coefficient are

100 W/°C super insulated home

250 W/°C average home

500 W/°C poorly insulated home.

4. DESCRIPTION OF PROGRAM OUTPUT

4.1 Thermal Analysis Results

The thermal analysis of the system is printed in monthly intervals with a yearly summary printed at the end. The results are an estimate of the system performance of a properly designed and installed system.

The definition of the six monthly output values follow.

SOLAR AVAIL. (GJ)

Total solar radiation incident on the collector over the time period in gigajoules.

SOLAR COLLECT (GJ)

Solar energy converted to heat by the solar collector over the time period.

SOLAR DELIVER (GJ)

Solar energy delivered to the load over the time period.

HEATING LOAD (GJ)

The total energy required to heat the air from the ambient temperature to the drier inlet temperature.

AUX. HEATING (GJ)

Auxiliary energy required to heat the air over the time period when the solar heating system is used.

FAN POWER (GJ)

The electrical energy required to operate the collector fan over the time period.

For System 2: MASS DRIED (KG) or TIME PERIOD (HOURS) is the total mass dried (based on dry weight) over the simulation period (if L1 = 0) or the time required to dry the mass (if L6 = 0).

For System 3: MAX TEMP (C) is the maximum temperature of the building over the time period.

After the monthly totals have been printed, the yearly sum of the six quantities is printed. A summary of energy use and savings is then printed in gigajoules.

The ENERGY SAVING DUE TO SOLAR is the percent reduction in auxiliary energy use attributable to the addition of the solar heating system i.e. percent solar. It is defined as:

$$\text{Energy Saving (\%)} = \frac{100\ (\text{Solar Energy Delivered} - \text{Fan Power})}{\text{Heating Load}}$$

AVERAGE SOLAR SYSTEM COLLECTION EFFICIENCY (%) is the solar contribution (solar collected minus fan power) divided by the solar radiation over the simulation period. It is important to note that if the daily air flow schedule is very short this value will be low regardless of the collector performance curve.

SOLAR CONTRIBUTION PER SQUARE METRE is the solar contribution divided by the collector area over the simulation period. A good application of a solar heating system would have a value of over 1.0 GJ/m²/yr.

For System 2, two warnings may be printed at the end of the simulation.

DRIER COULD NOT MEET DRYING LOAD DURING TIME SPECIFIED ON __ DAYS. This warning applies only to constant drying quantity systems (L1≠0). It is the total number of days that the drier cannot dry the required quantity by midnight.

DRIER COULD NOT DRY MATERIAL TO VALUE OF L4 FOR __ HOURS. This is the number of hours that the outlet moisture content of the material cannot achieve the value of L4. For these hours it is assumed that the drier does not operate.

5. PROGRAM ALGORITHM

5.1 Overview of Program Operation

The SIMAIR computer program calculates the performance of three types of air-based solar heating systems on an hour-by-hour basis. The basic assumption of the program is that for the purposes of calculating performance all variables, including solar radiation, ambient temperature and heating load can be considered constant for each hour. Thus, if the heating load varies widely within a given hour, the SIMAIR program is not applicable. The algorithm and flow chart for each type is discussed below. The algorithm for converting horizontal solar radiation data to that on the tilted surface is the same for all systems and is described in Section 5.4. Also common to all systems is the adjustment of the collector thermal performance characteristics from test conditions to actual conditions. The algorithm for performing this correction is given in Section 5.2.

5.1.1 Solar Preheated Ventilation Air Systems

The program calculation flow chart is shown in Figure 4. When the solar radiation and the input parameters are correct, the program begins the simulation. A new value of solar radiation and ambient temperature is read for each simulated hour and the process or ventilation air heating load is calculated (Q_L). If the load is zero the program goes to the next hour. If the solar radiation is zero the load is added to the previous load values and the program moves to the next hour.

If the solar radiation and heating load are greater than zero, F_R is calculated for that hour's flow rate. The equations for F_R are given in Section 5.2. The maximum solar contribution is calculated with the equation:

$$Q_S = F_R \cdot (\tau\alpha)_e \cdot A \cdot (K_b H_{bT} + K_d H_{dT} + K_r H_{rT}) \qquad (5.1)$$

where H_{bT}, H_{dT}, H_{rT} are the beam, diffuse and reflected solar radiation components on the tilted surface.

K_b, K_d, K_r are the beam, diffuse and reflected incident angle modifiers. The values are found by interpolating

Figure 4: Flow Chart for Solar Preheated Ventilation Air Systems

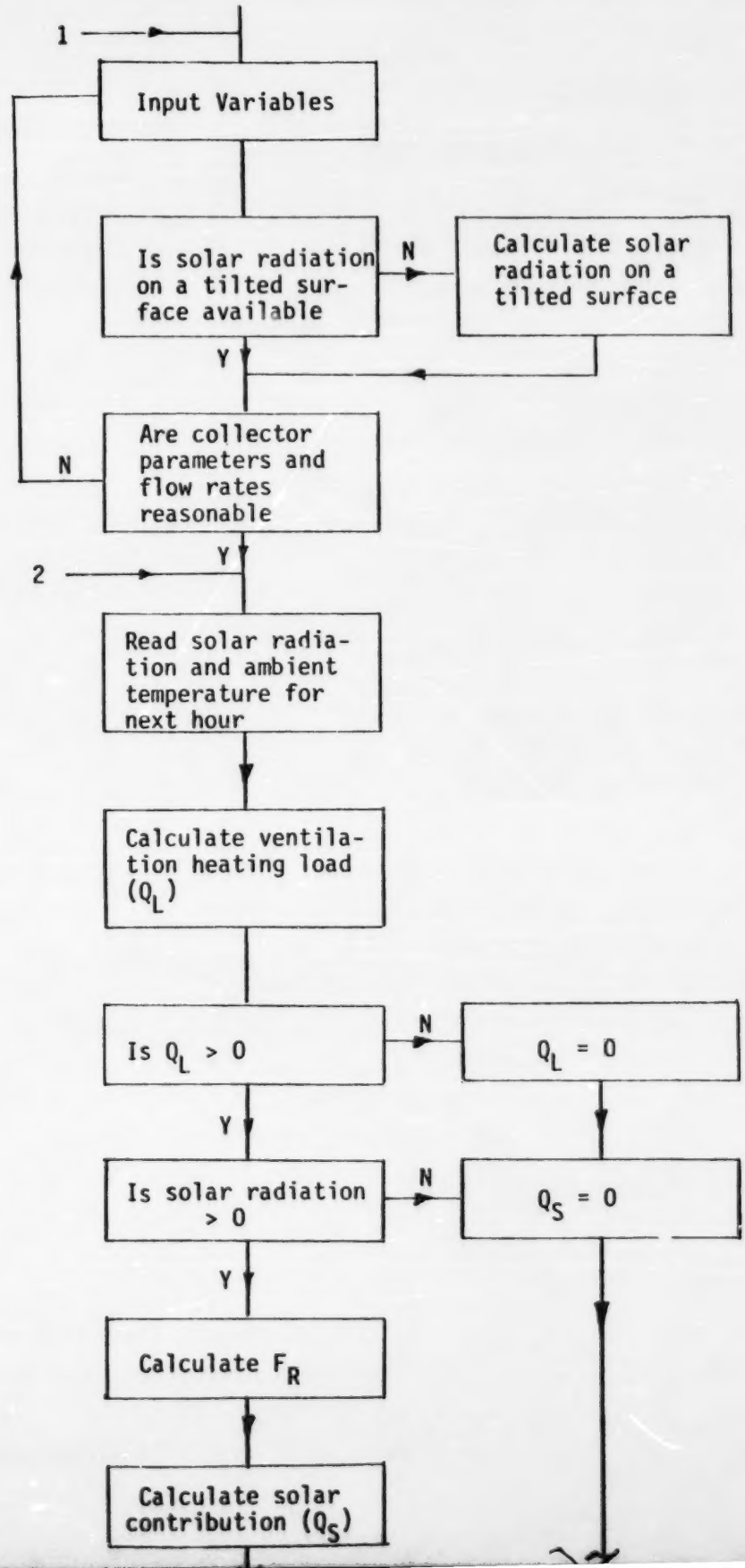
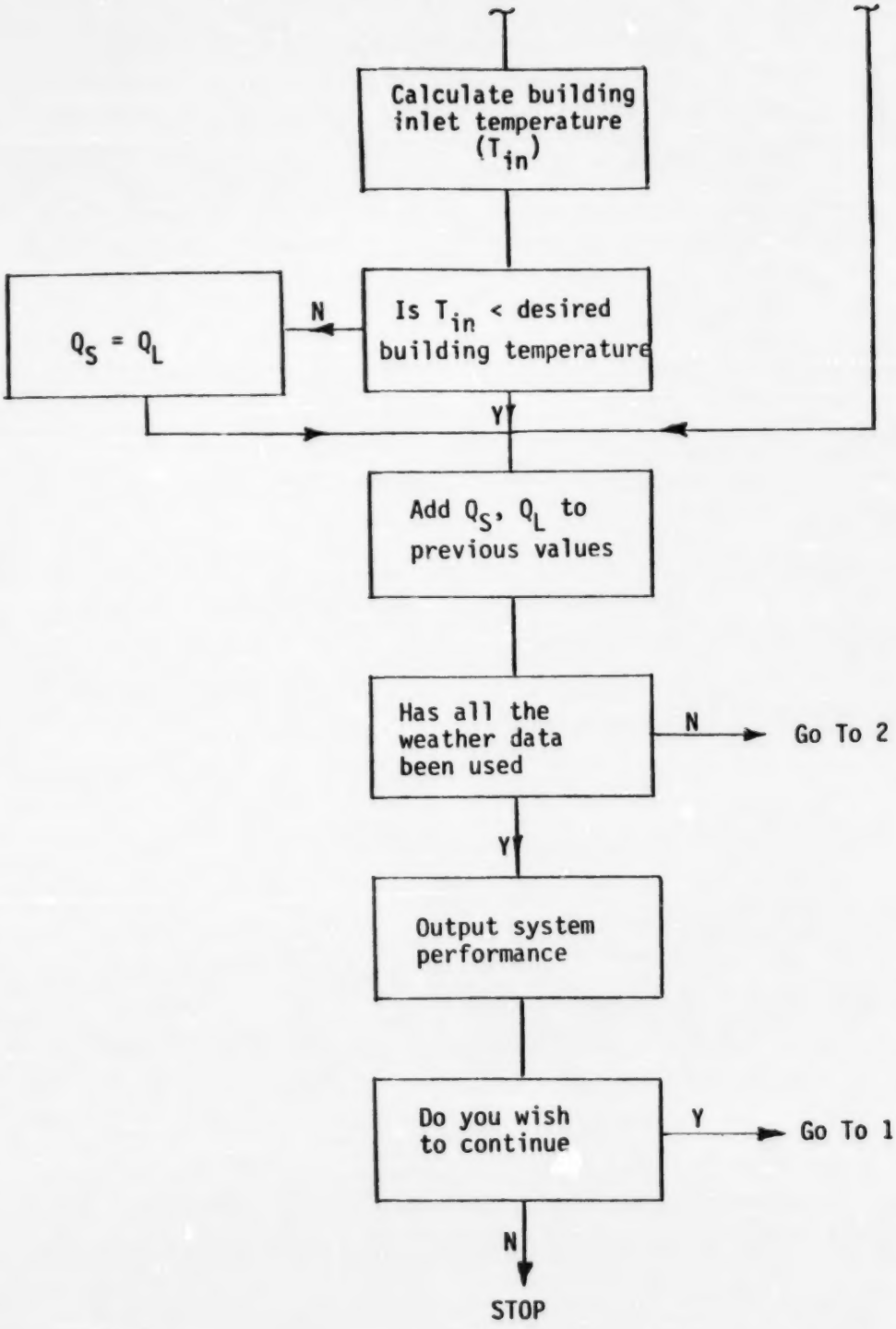


Figure 4 (Cont'd)



between the incidence angle values input by the user. The beam angle is calculated as shown in Section 5.4. The diffuse and reflected angles are calculated as recommended in reference 3.

The inlet temperature to the building (T_{in}) is calculated based on Q_S.

$$T_{in} = T_a + Q_S/(\dot{m}\ C_p) \tag{5.2}$$

If this temperature is greater than the desired temperature T_{bg}, then Q_S is made equal to Q_L. The values of Q_S and Q_L are added to the previous values and the program moves to the next hour.

At the end of each simulated month, the monthly totals of solar contribution and heating load are printed.

5.1.2 Crop and Material Solar Drying Systems

The program calculation flow chart is shown in Figure 5. After the solar radiation and input values are correct, the next step is to calculate the mass transfer coefficient for the drier based on the drier test data. This is an interative procedure requiring making an estimate of the mass transfer coefficient and calculating the drier outlet air condition. The mass transfer coefficient is subsequently altered until the calculated outlet condition equals the test outlet condition. The algorithm for estimating the drier outlet condition from the inlet condition is given in Section 5.3.

When the solar radiation and input parameters are correct, the program begins the simulation. The first step is to calculate the properties of the outside air: humidity ratio and enthalpy. The humidity ratio (W) is given by:

$$W_{ai} = \frac{W_{s_w}\ h_{fg_w} - C_{p_a}\ (Ta_d - Ta_w)}{C_{p_w}\ (Ta_d - Ta_w) + h_{fg_d}} \tag{5.3}$$

Figure 5: Flow Chart for Crop and Material Solar Drying Systems

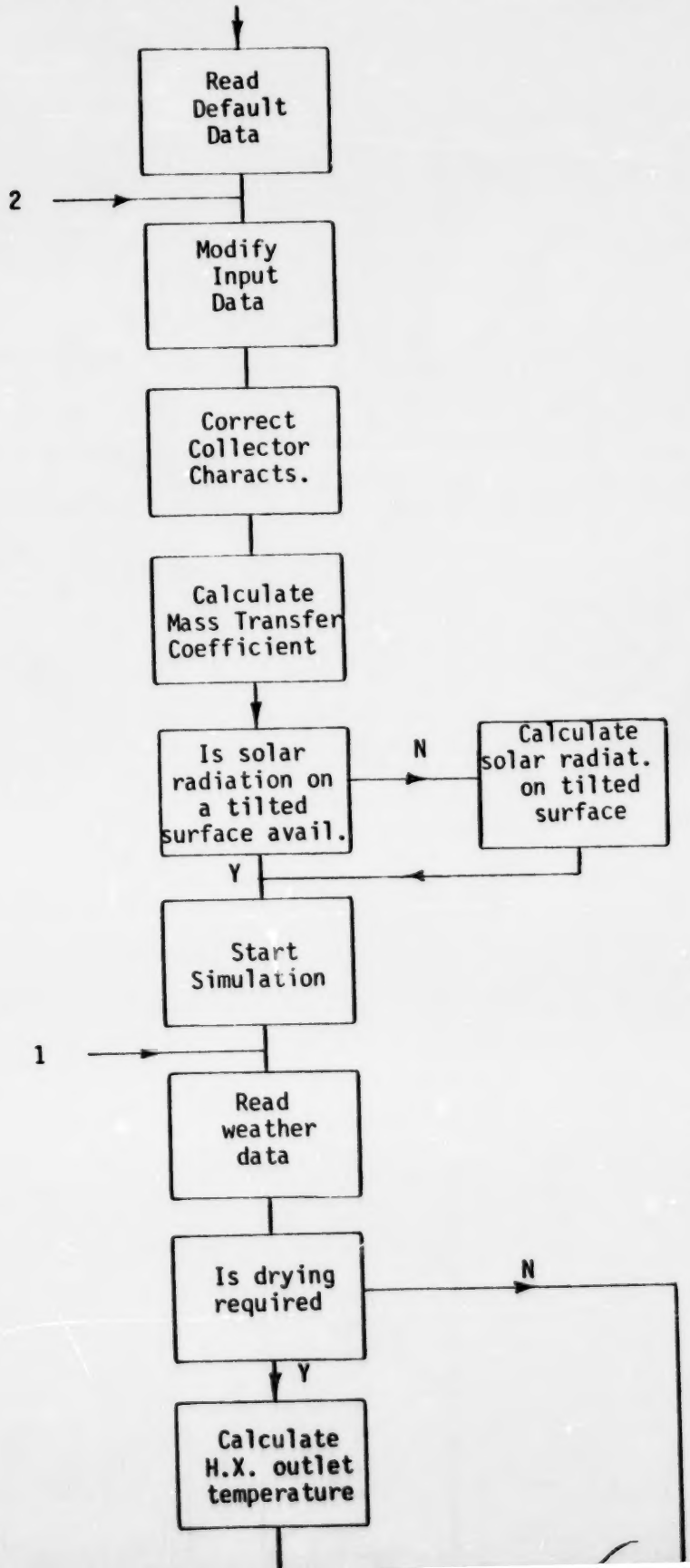
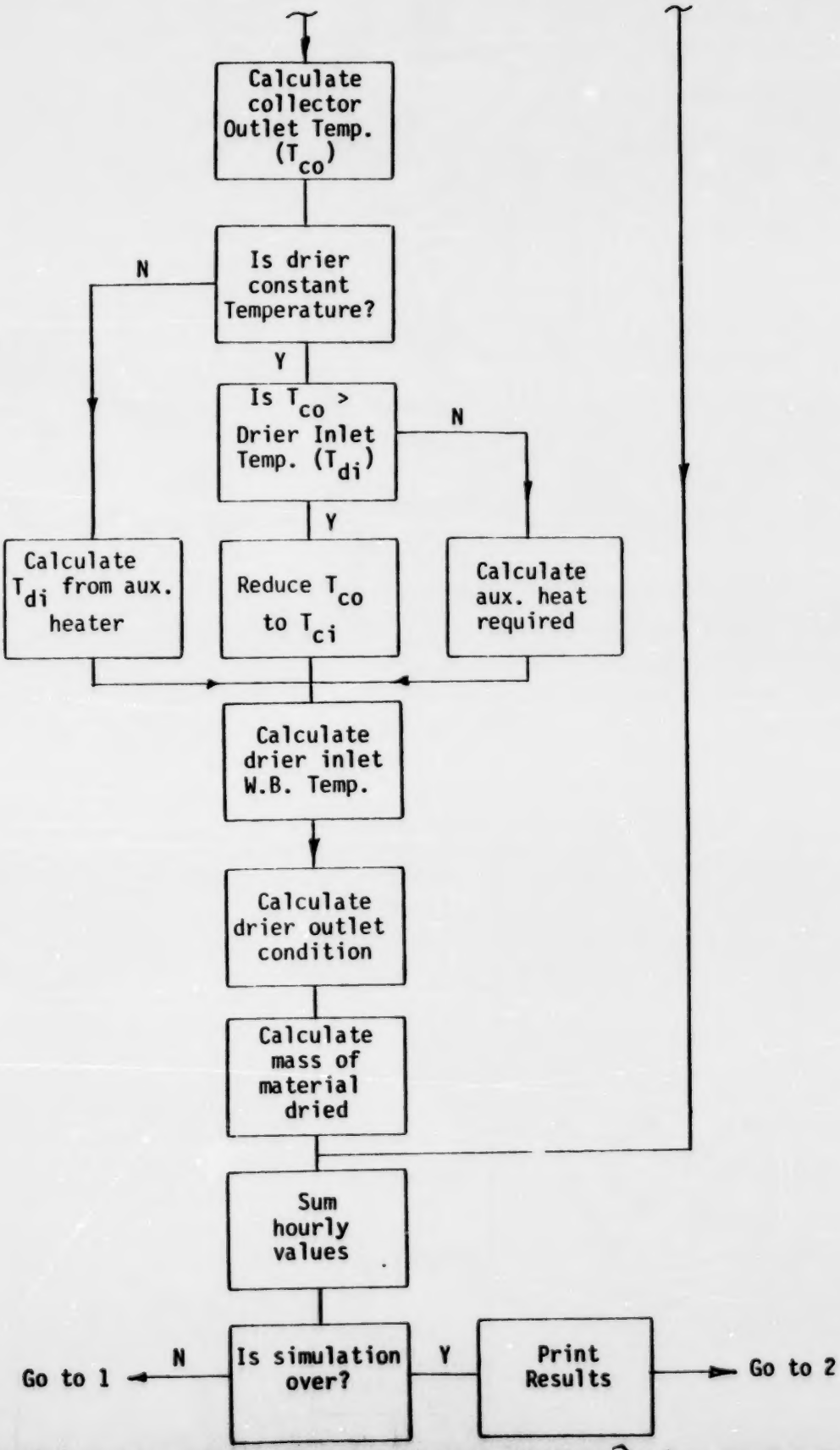


Figure 5 (Cont'd)



where Ta_d is the dry bulb ambient temperature,
Ta_w is the wet bulb ambient temperature,
C_{p_a} is the specific heat of air,
C_{p_w} is the specific heat of water,
h_{fg_w} is the latent heat of vapourization at the wet bulb temperature,
h_{fg_d} is the latent heat of vapourization at the dry bulb temperature,
W_{s_w} is the saturated humidity ratio at the wet bulb temperature given by: $\exp(-5.587+0.0761\ Ta_w-0.000474\ Ta_w^2+3.93\times10^{-6}\ Ta_w^3)$

(5.4)

The enthalpy of the air is:

$$h_{ai} = C_{p_e}\ Ta_d + 2502.3\ W_{ai}\ (KJ/kg) \tag{5.5}$$

where C_{p_e} is the effective specific heat of the air

$$= C_{p_a} + C_{p_w}\ W_{a_i} - 2.425\ W_{a_i} \tag{5.6}$$

If a heat exchanger is used before the collectors, the next step is to calculate the increase in air inlet temperature.

$$T_{hx_o} = T_{a_i} + E_{hx}\ (T_{do} - T_{a_i}) \tag{5.7}$$

where E_{hx} is the effectiveness of the heat exchanger,
T_{hx_o} is the temperature out of the heat exchanger,
T_{do} is the temperature of the air leaving the drier (assumed to be equal to the drier inlet temperature).

The solar energy collected is calculated as:

$$Q_S = A_c\ [F_R\tau\alpha\ (K_b\ H_{bT} + K_d\ H_{dT} + K_r\ H_{rT}) - F_R U_L\ (T_{hx_o} - Ta_d)] \tag{5.8}$$

The collector outlet temperature is given by:

$$T_{co} = T_{hx_o} + Q_S / (C_{p_e} \dot{m}_a) \tag{5.9}$$

If the collector outlet temperature is greater than the maximum allowable temperature, it is assumed that some of the outside air bypasses the collector and is mixed with the collector outlet air. If a heat exchanger is placed after the collector the program calculates the change in air temperature using Equation 5.7 with T_{a_i} replaced with the collector outlet temperature. For the constant temperature drier, if the collector (or heat exchanger) outlet temperature is too cold, auxiliary heat is added to bring the air up to the desired temperature. For the constant heat input drier, the inlet temperature to the drier is:

$$T_{d_i} = T_{c_o} + Q_{heat} / (C_{p_e} \dot{m}_a) \tag{5.10}$$

If the drier inlet temperature exceeds the maximum allowable, the auxiliary heater is shut off.

The next step in the program is to calculate the wet bulb temperature at the drier inlet. This requires an iterative solution to Equations 5.3 and 5.4. The drier inlet wet bulb temperature is used to calculate the saturated humidity ratio using Equation 5.5.

When the drier inlet condition is known, the drier performance is calculated. The algorithm to estimate the drier outlet condition is given in Section 5.3. Knowing the amount of water that is removed, the quantity of material dried can be calculated:

$$\dot{m}_d = \frac{\dot{m}_a (Wa_o - Wa_i)}{(x_i - x_f)} \tag{5.11}$$

where $\dot{m}_d$ is the rate of material drying,
x_i is the initial moisture content of the material,
x_f is the desired final moisture content of the material.

The hourly values of the above energy and mass flows are summed and monthly and yearly totals printed.

5.1.3 Zero Storage Space Heating Systems

The program calculation flow chart is shown in Figure 6. When the solar radiation and the input parameters are correct, the program begins the simulation. The first step is to calculate the building heat loss for that hour:

$$Q_{bg} = UA_b \, (T_{bg} - T_a) - Q_{ig} \tag{5.12}$$

If there was no other energy input to the building, the building temperature would become:

$$T'_{bg} = T_{bg} - Q_{bg}/C_{bg} \tag{5.13}$$

Provided that this new value is below the maximum allowable temperature and heat can be collected, the solar heat delivered to the building can be calculated:

$$Q_S = [F_R\tau\alpha \, (K_b \, H_{bT} + K_d \, H_{dT} + K_r \, H_{rT}) - F_R U_L \, (T_{bg} - T_a)] \, A \tag{5.14}$$

The new building temperature is then:

$$T''_{bg} = T'_{bg} + Q_S/C_{bg} \tag{5.15}$$

If this building temperature is greater than the maximum allowable temperature, then all the solar heat for that hour is not usable. The amount of solar heat that is usable is given by:

$$Q'_S = Q_S \, (T_{max} - T'_{bg}) \, / \, (T''_{bg} - T'_{bg}) \tag{5.16}$$

Figure 6: Flow Chart for Zero Storage Space Heating System

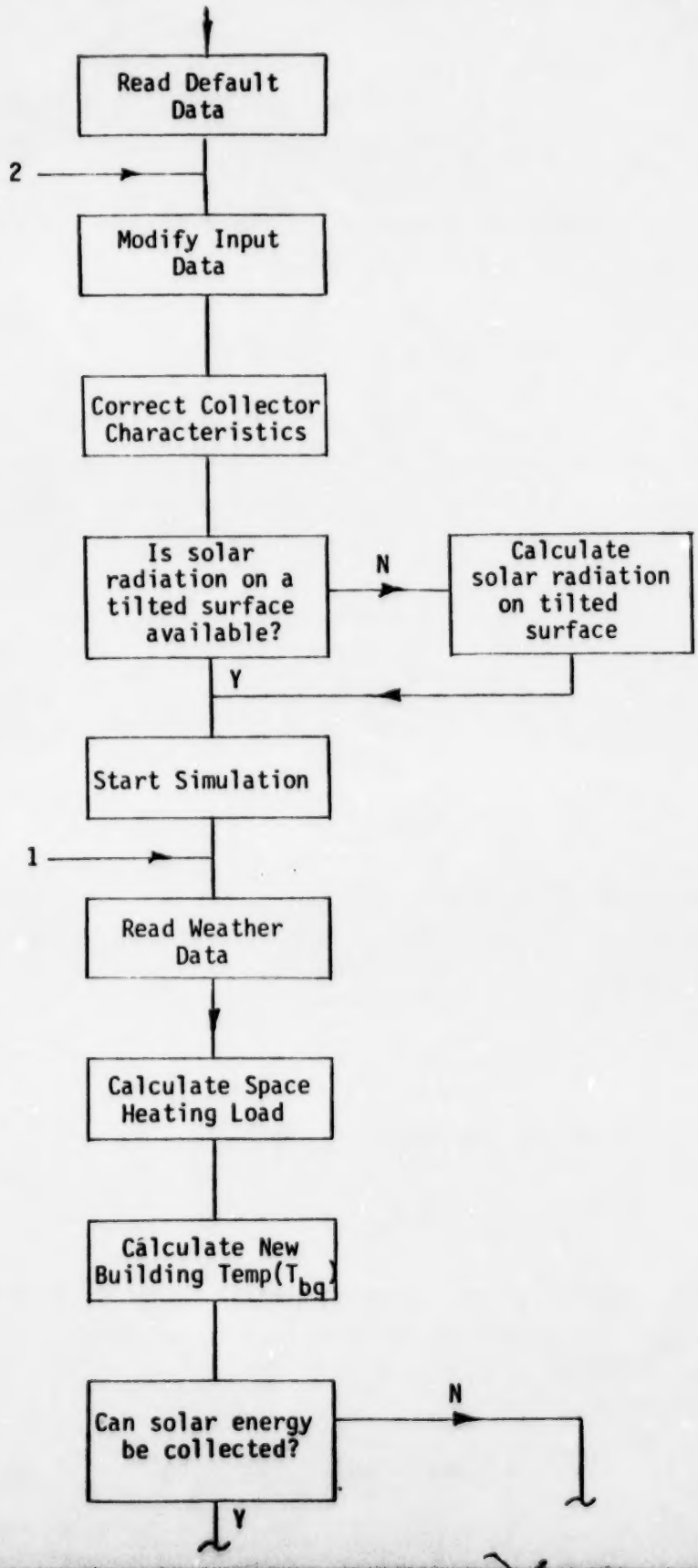
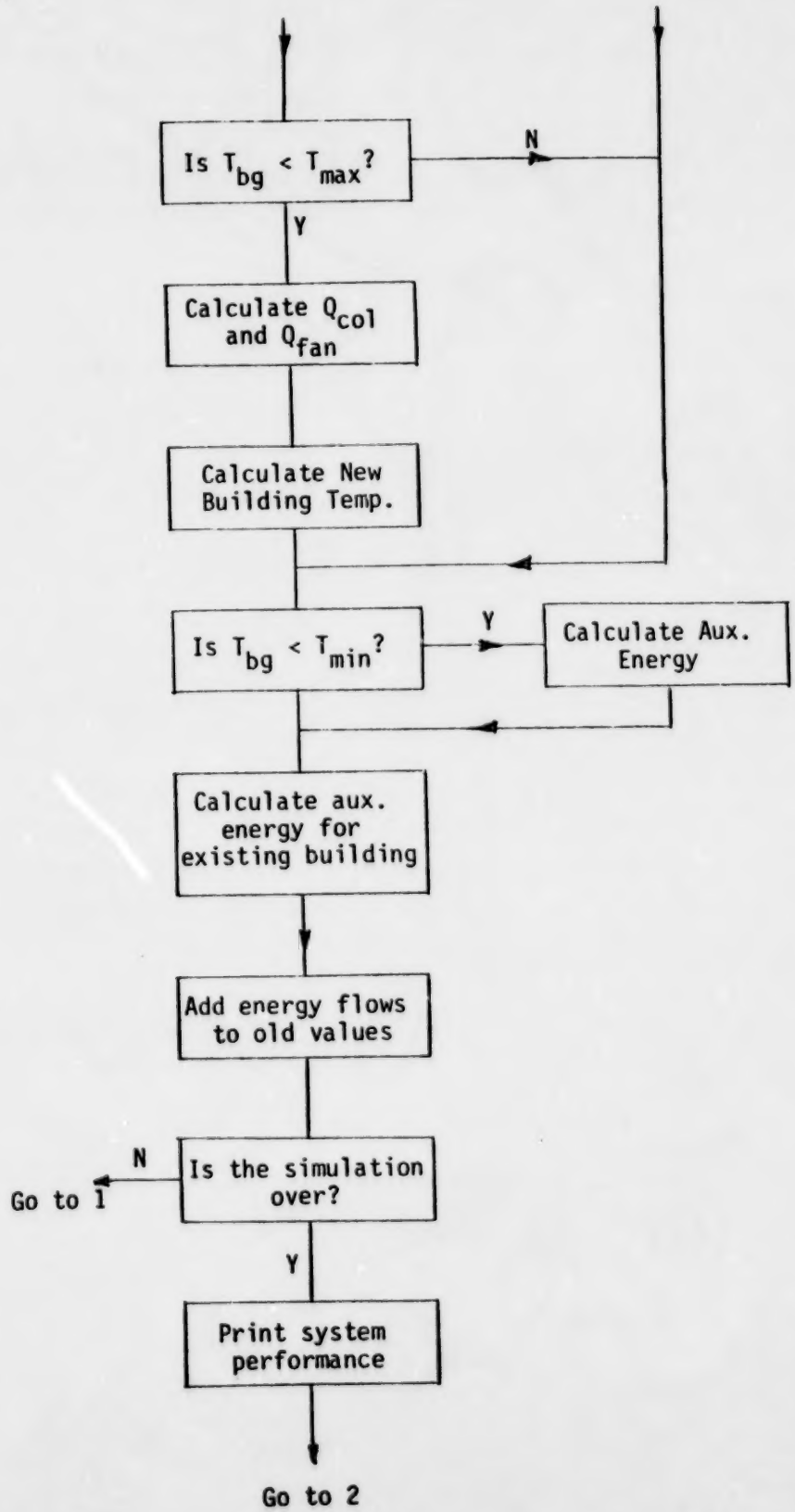


Figure 6 (Cont'd)



If the building temperature is below the minimum allowable temperature then auxiliary energy is used to raise the temperature.

The building heating load (i.e. how much auxiliary energy is required to heat the building without the solar heating system) is calculated in the same manner as above, but without any solar gains. It should be noted that the yearly building heating load is not the sum of the hourly building heat losses from the solar heated building. The solar building will be at a higher temperature and thus have higher heat losses than the building with no solar system. Since we are interested in the energy savings as a result of installing a solar heating system, our base energy consumption should be that of the existing or non-solar building.

The hourly values of the above energy flows are summed and monthly and yearly totals printed.

5.2 Algorithm for the Calculation of F_R

Two separate algorithms are used for the calculation of F_R, one for parallel plate and V-corrugated solar collectors and one for fibre matrix solar collectors. It is important to note that a correction on F_R for the number of collectors in series is not necessary provided that the mass flow per unit area is kept constant.

5.2.1 F_R for Parallel Plate and V-Corrugated Solar Collectors

F_R is given by:

$$F_R = \frac{\dot{m}\, C_p}{U_L} \left(1 - \exp(-F' U_L / (\dot{m}\, C_p)\right)$$

where $F' = U_o/(U_o + U_L)$

All of the variables are constant except for $\dot{m}$ (mass flow rate per unit area) and U_o (the total plate to fluid heat transfer coefficient). If $\dot{m}$ changes each hour then a new value of U_o must be calculated for each hour. U_o depends on the collector design.

i) U_o for flow below the absorber plate

$$U_{o_b} = R_A\, h_{pf} + h_{pf2}\, h_r/(h_f + h_{pf2})$$

where R_A is the ratio of absorber area to collector aperture area (typically 1 for parallel plate and greater than 1 for V-corrugated),

h_{pf} is the convective heat transfer coefficient between the top absorber plate and the air stream,

h_{pf2} is the convective heat transfer coefficient between the bottom absorber plate and the air stream,

h_r is the radiative heat transfer coefficient between the upper and lower absorber plates.

If we assume that the inside of the air channel is painted black:

$$h_r = \sigma(T_1^2 + T_2^2)\,(T_1 + T_2)\,(1.111 + 0.111/R_A)$$

where T_1 and T_2 are the temperatures of the upper and lower absorber plates.

ii) U_o for flow above the absorber plate

$$U_{o_a} = h_{pf2} + h_{pf}\, R_A + \frac{h_{pf}\, R_A\, U_L}{h_r} + \frac{h_{pf}\, R_A\, h_{pf2}}{h_r}$$

where h_{pf} is as defined above,

h_{pf2} is the convective heat transfer coefficient between the glazing and the air stream,

h_r is the radiative heat transfer coefficient between the absorber and the glazing.

iii) U_o for flow above and below the absorber plate

$$U_{o_{ab}} = U_{o_a} + U_{o_b}$$

Calculation of h_{pf}

The value of h_{pf} depends on the construction of the absorber (or glazing). The program can estimate h_{pf} for flat and V-corrugated surfaces as follows.

The equations for h_{pf} given below were taken from:

Hollands, K.G.T. and Shewen, E.C., Journal of Solar Energy Engineering, Vol. 103, No. 4, November, 1981. "Optimization of Flow Passage Geometry for Air-Heating, Plate-Type Solar Collectors".

i) h_{pf} for flat plates

$$h_{pf} = 4Nu\ k\ (R_A + 1)/4b$$

where k is the conductivity of air,
b is the spacing between the upper and lower absorber plates,
Nu is the Nusselt number.

for Re < 2000 (Reynolds number)

$$Nu = 5.385 + 0.148\ Re\ b\ /\ L$$

L is the collector flow path length.

for 2000 < Re < 10000

$$Nu = 0.00044\ Re^{1.2} + 9.37\ Re^{0.471}\ b\ /\ L$$

for 10000 < Re < 100000

$$Nu = 0.03\ Re^{0.74} + 0.788\ Re^{0.74}\ b\ /\ L$$

where $Re = \frac{2\ \dot{m}\ L}{\mu}$

if Re is greater than 100000 the program sets Re equal to 100000.

ii) h_{pf} for V-corrugated plates

$$h_{pf} = \frac{Nu\ k\ (R_A + 1)}{4\ b}$$

for Re < 2800

$$Nu = 2.821 + 0.126\ Re\ b\ /\ L$$

for 2800 < Re < 10000

$$Nu = 1.9 \times 10^{-6}\ Re^{1.79} + 225\ b\ /\ L$$

for 1000 < Re < 100000

$$Nu = 0.0302\ Re^{0.74} + 0.242\ Re^{0.74}\ b\ /\ L$$

5.2.2 F_R for Fibre Matrix Collectors (see Figure 7)

The equation for F_R is as given for parallel plate collectors, F', however, is defined as:

$$F' = \frac{h_r\ h_{pf} + U_o\ U_L + U_o\ h_r + h_{pf}\ U_o}{(U_L + h_r + h_{pf})\ (U_o + h_r) - h_r^2}$$

where h_{pf} is the convective heat transfer coefficient between the lower glazing and the air stream,

h_r is the radiative heat transfer coefficient between the fibre matrix and the lower glazing. Because of the nature of the fibre matrix, h_r is a weighted average of the value of h_r for each layer of fibres with the top layer having a weighting of 1 and the bottom layer having a weighting of 0. By integrating the values of h_r over the depth of the matrix, h_r can be approximated by:

$$h_r = h_{r1}/3 + h_{r2}/6$$

where $h_{r1} = \sigma(T_1^2 + T_2^2)\ (T_1 + T_2)/1.222$

$h_{r2} = \sigma(T_{fo}^2 + T_2^2)\ (T_{fo} + T_2)/1.222$

T_1 is the temperature of the top of the fibre matrix,

T_2 is the temperature of the lower glazing.

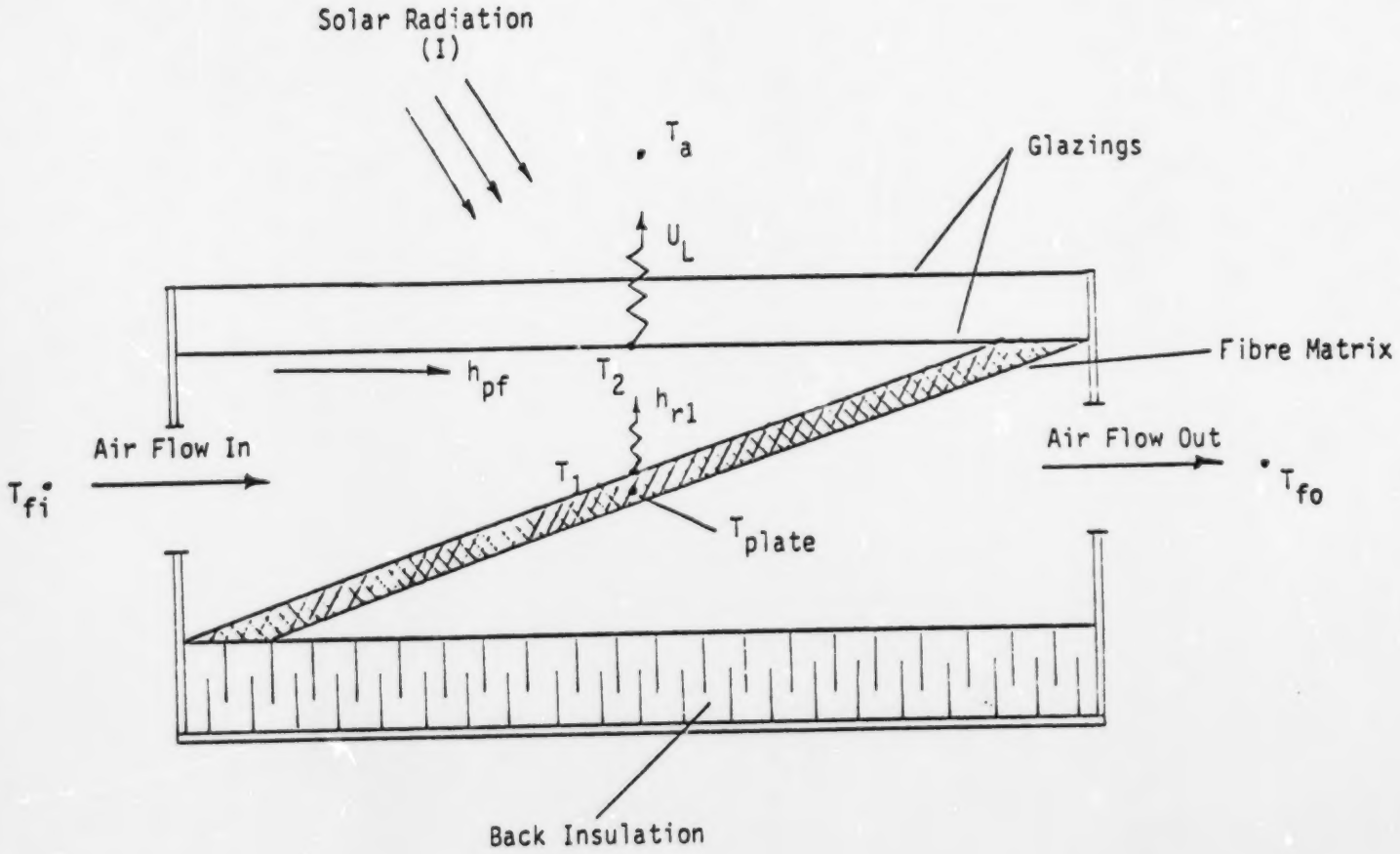


Figure 7: Schematic of Fibre Matrix Solar Collector

The average fibre matrix plate temperature can be calculated by:

$$T_{plate} = T_{fm} + Qu/U_o$$

where U_o is the fibre matrix to air heat transfer coefficient,
T_{fm} is the average fluid temperature.

Estimating U_o for fibre matrix collectors is much more difficult than for parallel plate collectors. The reference used for these equations is:

Kays, W. and London, A.L., Compact Heat Exchangers, Second Edition, McGraw-Hill, New York, 1964, pg. 129.

Kays and London give equations of the form for fibre matrices:

$$U_o = x \left(\frac{A_{fm}}{A_c}\right) \dot{m} C_p/(Re^y Pr^{2/3})$$

where Pr is the Prandtl number,
x and y are constants dependent on the type and shape of the matrix.

For a similar type of fibre matrix Kays and London give:

$x = 1.3 \qquad y = 0.45$

By knowing the average fibre matrix diameter and the fibre matrix mass, the ratio of heat transfer to collector area can be calculated. For the Amherst collector:

$$\frac{A_{fm}}{A_c} \cong 80.$$

5.3 Algorithm to Calculate Drier Performance

The program can simulate the performance of two types of driers: counter-flow and cross-flow. The theory of drying and the two types of driers are described below.

i) Drying Theory

The drying theory used in the program is based on the analysis by Threlkeld (2). Drying rate (R) is:

$$R = h_D (W_{s_w} - W)$$

where h_D is the mass transfer coefficient,
W_{s_w} is the saturated humidity ratio at the wet bulb temperature,
W is the humidity ratio of the air.

The mass transfer coefficient depends on many factors, however, the most important factor is the moisture content of the material. A typical curve for mass transfer coefficient versus material moisture content is shown in Figure 8. The mass transfer coefficient (and therefore the drying rate) is constant for moisture contents above a critical moisture content. At the critical moisture content, the rate of water migration from inside the material to the surface equals the rate of water evaporation. Below the critical moisture content, the mass transfer coefficient decreases as the moisture content decreases because water cannot be supplied fast enough to the material surface. The rate continues to decrease until the equilibrium moisture content is reached. At this point the drying rate is zero. Some researchers have suggested a drying curve with two regions (with different slopes) below the critical moisture content. It is unlikely, however, that the designer would have information on the drying rate for the two regions. Thus, for simplicity a single region drying curve is used by the program.

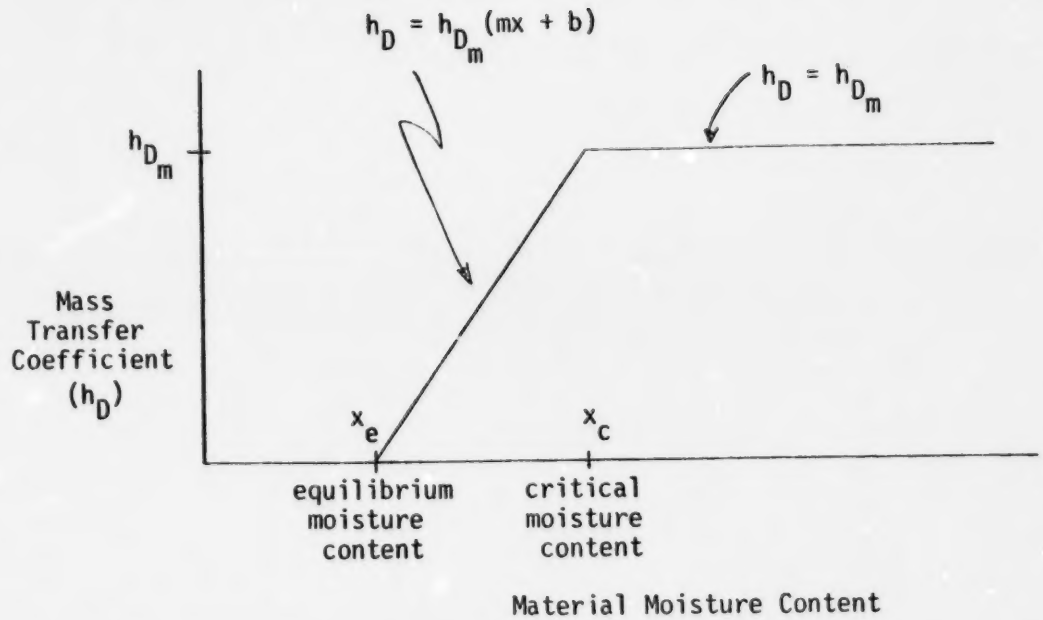


Figure 8: Typical Relationship between Mass Transfer Coefficient and Material Moisture Content

ii) Counter-Flow Drier

In a counter-flow drier the material and air move in parallel but opposite directions. The advantage of this design is that the driest air is in contact with the driest material. This arrangement allows the maximum amount of water to be transferred to the air.

To determine the performance of this drier, the drying rate equation must be integrated over the length. In the drying rate equation h_D and W are changing as the air flows through the drier. This can be written as:

$$\dot{m}_a dW = R\, A_D\, M_D\, dL$$

where A_D is the drying area of the material,
M_D is the mass of the material,
L is the length of the drier.

This equation can be re-written as:

$$dL = \frac{\dot{m}_a\, d\, W}{A_D\, M_D\, h_D\, (W_{s_w} - W)}$$

The drier can be split into two regions: constant h_D and $h_D = h_{D_m}\,(mx + b)$, where h_{D_m} is the constant value of h_D above x_c (critical moisture content). The drier length can be split into these two regions.

$$L = L_1 + L_2$$

Constant h_D:

$$L_1 = \frac{\dot{m}_a}{A_D\, M_D\, h_{D_m}} \int_{W_c}^{W_o} \frac{d\, W}{(W_{s_w} - W)}$$

Variable h_D:

$$L_2 = \frac{\dot{m}_a}{A_D M_D h_{D_m}} \int_{W_i}^{W_c} \frac{d W}{(W_{s_w} - W)(mx + b)}$$

where $m = 1./(x_c - x_e)$

$b = -x_e/(x_c - x_e)$

where x_e is the equilibrium moisture content.

The solution to these equations is:

$$L_1 = \frac{\dot{m}_a}{A_D M_D h_{D_m}} \ln \left(\frac{W_{s_w} - W_c}{W_{s_w} - W_o} \right)$$

$$L_2 = \frac{}{A_D M_D h_{D_m}} Z_1 \left(\ln \left(\frac{Z_2 - Z_1}{Z_2 + Z_1} \right) - \ln \left(\frac{Z_3 - Z_1}{Z_3 + Z_1} \right) \right.$$

where $Z_1 = (a_2^2 - 4a_1a_3)^{\frac{1}{2}}$

$Z_2 = 2a_3W_c + a_2$

$Z_3 = 2a_3W_i + a_2$

$a_1 = (mx_f - m \dot{m}_a W_i/\dot{m}_D + b) W_{s_w}$

$a_2 = m \dot{m}_a (W_{s_w} + W_i)/\dot{m}_D - mx_f - b$

$a_3 = - m \dot{m}_a/\dot{m}_D$

This equation must be solved iteratively for W_o and $\dot{m}_D$. The quantity $L A_D M_D h_D$ is a constant determined from the test results.

Most materials that are dried are relatively small in diameter (e.g. grains). The small diameter means that there is a large area for heat transfer. Thus, like a rock bed thermal storage unit, the air flowing through the drier becomes very close to the temperature of the material it is flowing through. For a counter-flow drier, this results in a stratified container with the grain in the bottom (i.e. inlet) half equal to the inlet air temperature and the air at the top (i.e. outlet) half equal to the grain temperature.

If the grain temperature is not the same as the air inlet temperature, the air outlet temperature is reduced to the grain inlet temperature. The humidity ratio of the air is also reduced so as not to exceed 100% relative humidity at the new outlet temperature. Since most of the drying (i.e. water removal) occurs in the inlet half of the drier, the drying rate is not significantly affected by the grain inlet temperature.

iii) Cross-Flow Drier

In a cross-flow drier the material and air flow in perpendicular directions. For the purposes of analysis the drier can be divided into two regions: constant mass transfer coefficient and variable mass transfer coefficient. The analysis of the constant mass transfer coefficient region is the same as for the counter-flow drier. For the variable region:

$$\frac{A_D M_D}{L \dot{m}_a} = \int_{W_i}^{W_{o2}} \frac{d W}{h_D (W_{s_w} - W)}$$

Unlike the counter-flow drier, however, h_D does not vary in the flow direction. If an average h_D is used in the above equation, then the solution is the same as for the constant region.

The drier outlet humidity ratio is the weighted average of the outlet humidity ratio from the two drying regions (constant and variable h_D).

$$W_o = W_{o1} X + W_{o2} (1 - X)$$

where $X = Z/(1 + Z)$

$$Z = \frac{(x_i - x_c)(W_{02} - W_i)}{(x_c - x_o)(W_{o1} - W_i)}$$

For cross-flow driers, if the grain inlet temperature is lower than the drier air inlet temperature, the drying rate is reduced. The reduction in drying rate is a result of a lower average air temperature and thus a lower water carrying capacity in the drier. To calculate the drying rate, the average drier air temperature based on the air and material flow rates is used.

5.4 Algorithm to Correct for Collector Loop Air Leakage

All ductwork leaks air. While air leakage is not critical for indoor ductwork, it can seriously decrease performance for outdoor ductwork. The collector performance characteristics can be modified to account for this decrease in performance. The algorithm used in the program is taken from reference 1 with extensions for the case of air leaking out of the collector loop. It will be assumed that air leaks are evenly divided between the ductwork before the collector and the ductwork after the collector.

There are four possible cases for air leakage: air leaking in before collector, air leaking in after collector, air leaking out before collector, and air leaking out after collector. Each of these cases is discussed below.

(i) Air Leaking In Before Collector

In this case air leaks into the collector and out of the building. The extra heat lost out the building can be included in the $F_R U_L$ term as follows:

$$F_R U_{L_{leaks}} = F_R U_L (1 - L_R) + \frac{L_R \dot{m} Cp}{A_c}$$

where L_R is the ratio of the leakage rate to the collector flow rate.

In this case the full air flow goes through the collector so there is no change in the F_R or the $F_R\tau\alpha$ term.

(ii) Air Leaking In After Collector

This case is similar to case (i) except that some of the air does not flow through the collector. Thus the collector heat removal factor must be ajusted for this lower flow rate. For small changes in mass flow rate:

$$F_R' = F_R \left(\frac{1 - L_R}{X}\right)\left[1 - (1 - X)^{\frac{1}{1 - L_R}}\right]$$

$$\text{where } X = \frac{F_R U_L A_c}{\dot{m}\, Cp}$$

Thus the collector characteristics are:

$$F_R U_{L_{leaks}} = F_R U_L \left(\frac{F_R'}{F_R}\right) + \frac{L_R \dot{m}\, Cp}{A_c}$$

$$F_R\tau\alpha_{leaks} = F_R\tau\alpha\left(\frac{F_R'}{F_R}\right)$$

Combining cases (i) and (ii) gives the collector characteristics for air leakage before and after the collector:

$$F_R U_{L_{leaks}} = F_R U_L \left(\frac{F_R'}{F_R}\right)(1 - L_R) + 2L_R \frac{\dot{m}\, Cp}{A_c}$$

$$F_R\tau\alpha_{leaks} = F_R\tau\alpha\left(\frac{F_R'}{F_R}\right)$$

where L_R is one half the ratio of the air leakage rate to collector flow rate and F_R' is calculated based on one half the leakage rate.

(iii) Air Leaking Out Before Collector

If air leaks out of the collector loop, then air must leak into the building. In this case a reduced flow rate goes through the collector and the air exhausted to the outdoors is at room temperature. This case has the same effect on collector characteristics as in case (ii).

(iv) Air Leaking Out After Collector

This case is similar to case (i) except that the air exhausted to the outdoors is at the collector outlet temperature as opposed to the building temperature. The collector characteristics for this case are:

$$F_R U_{L_{leaks}} = F_R U_L \ (1 - L_R) + \frac{\dot{m}\ Cp\ L_R}{A_c}$$

$$F_{R\ leaks} = F_R \ (1 - L_R)$$

Combing cases (iii) and (iv) gives the case of equal air leaks out before and after the collector.

$$F_R U_{L_{leaks}} = F_R U_L \ (\frac{F'_R}{F_R}) \ (1 - L_R) + \frac{2L_R\ \dot{m}\ Cp}{A_c}$$

$$F_{R\ leaks} = F_R \ (\frac{F'_R}{F_R}) \ (1 - L_R)$$

where L_R is one half the ratio of the air leakage rate to collector flow rate and F'_R is calculated based on one half the leakage rate.

5.5 Algorithm to Process Weather Data

Most Canadian weather stations measure only total solar radiation on a horizontal surface. Most solar collectors, however, are tilted towards the sun to increase the incident solar radiation. The program determines hourly values of total solar radiation (beam, diffuse and reflected) on a tilted surface and stores the values in a scratch file. The algorithm for converting horizontal solar radiation to tilted solar radiation is similar to the method used in "Solar Engineering of Thermal Processes" by Duffie and Beckman (3).

In order to estimate the solar radiation on a tilted surface it is necessary to split the total measured horizontal solar radiation into its two components: beam and diffuse. It is possible to estimate the amount of diffuse solar radiation from the ratio of the measured solar radiation to the extraterrestrial solar radiation. If this ratio is low then the solar radiation must be mostly diffuse; if this ratio is high the solar radiation must be mostly beam.

When the beam and diffuse solar radiation components are known, standard geometric relations can be used to estimate the solar radiation components on a tilted surface. When estimating solar radiation on a tilted surface a third component is introduced: reflected radiation. Reflected radiation can be estimated from the beam radiation and the ground albedo or reflectivity.

The program equations and execution procedure are given below.

At the start of each day the solar constant and the earth's solar declination are calculated. The solar constant is given by:

$$S_c = 4871.0\ (1. + 0.33\ \cos(2\pi N/365) \qquad \text{in KJ/(hr m}^2\text{)}$$

where N is the day number (Jan 1 is 1).

The earth's declination is given by:

$$\delta = \frac{23.45 * 2\pi}{360} \sin\left(\frac{2\pi}{360}(284 + N)/365\right) \qquad \text{(in radians)}$$

These values are assumed constant for each day. All other calculations are made on an hourly basis.

The first step is to read the measured weather values from the data file. For each hour the weather data file contains six values in the following order.

1) month number (1-12)
2) day number (1-31)
3) hour number (1-24)
4) ground reflectivity
5) solar radiation on a horizontal surface (in Watts/m^2)
6) ambient temperature (in °C)

The extraterrestrial solar radiation on a horizontal surface is calculated by:

$$H_{ex} = S_c \cos(\theta_z)$$

where $\cos(\theta_z)$ is cosine of the zenith angle (angle between the beam and the vertical),

$\cos(\theta_z) = \cos(\phi) \cos(\delta) \cos w + \sin\phi \sin\delta$

ϕ is the latitude of the location,

w is the hour angle.

The diffuse solar radiation (H_d) can be estimated using a correlation by Orgill and Hollands (4).

$$H_d = 0.1769\ H \qquad \text{if } 0.75 < K_T$$

$$H_d = (1.55699 - 1.84013\ K_T)\ H \qquad \text{if } 0.35 \leq K_T \leq 0.75$$

$$H_d = (1. - 0.248857\ K_T)\ H \qquad \text{if } 0.0 \leq K_T \leq 0.35$$

where H is the measured hourly solar radiation,

K_T is the ratio of measured solar radiation to the extra-terrestrial solar radiation,

$$K_T = H / H_{ex}$$

The beam radiation (H_b) is simply the total measured solar radiation minus the diffuse radiation.

$$H_b = H - H_d$$

The next step is to calculate the ratio of beam radiation on the tilted surface to that on the horizontal surface (R_b).

$$R_b = \cos(\theta_T) / \cos(\theta_z)$$

where $\cos(\theta_T)$ is the cosine of the angle of incidence of beam radiation, between the beam and the normal to the surface,

$$\cos(\theta_T) = [\sin(\delta)\ \sin(\phi)\ \cos(s)] - [\cos(\delta)\ \cos(\phi)\ \cos(s)\ \cos(w)] + [\cos(\delta)\ \sin(\phi)\ \sin(s)\ \cos(\gamma)\ \cos(w)] + \cos(\delta)\ \sin(s)\ \sin(\gamma)\ \sin(w)$$

γ is the azimuth angle measured from south (east is positive, west is negative).

Thus the beam solar radiation on the tilted surface is:

$$H_{bT} = R_b\ H_b$$

The diffuse solar radiation component on the tilted surface is estimated using the radiation view factor from the collector to the sky with correction factors for non-uniform distribution of diffuse radiation.

The correction factors for anisotropic diffuse radiation are taken from Temps and Coulson (5) and Klucher (6). The resulting equation is:

$$H_{dT} = \left(\frac{1 + \cos(s)}{2}\right)(1 + F \sin^3(s/2))(1 + F \cos^2(\theta_T) \sin^3(\theta_z)) H_d$$

where $F = 1 - (H_d/H)^2$

The reflected solar radiation on the tilted surface (H_r) is:

$$H_r = \left(\frac{1 - \cos(s)}{2}\right) \rho H$$

where ρ is the ground reflectivity.

The total solar radiation on the tilted surface (H_T) is the sum of the beam,diffuse and reflected solar radiation components.

Hourly values of beam, diffuse and reflected solar radiation on a tilted surface, ambient temperature, and θ_T are written to the scratch file. When all the data have been processed and written to the scratch file, the file is rewound to be ready for the system simulation.

6. PROGRAM SETUP

Six file definitions must be made before the program can be run:

Terminal (Read) - (Unit 5) is the device used for data input. The program will send all questions and prompts to this device.

Printer - (Unit 7) is the device that receives the printed output (i.e. a printer). If a send and receive printer is being used, Unit 7 need not be defined.

Weather Data - (Unit 2) is the file containing the TRNSYS compatible weather data. The data must be written in the format (2X, I2, 2X, I2, 2X, I2, F3.1, I3, F6.1) and contain month number, day number, hour number, ground reflectivity, ambient temperature (°C), and solar radiation on a horizontal surface (W/m^2).

Wet Bulb Data - (Unit 1) is the file containing the wet bulb temperatures. The data must be written in the Atmospheric Environment Service standard format (185 characters per line).

Processed Data - (Unit 9) is the file that is created by the program containing the solar radiation on the tilted surface and ambient temperature.

Terminal (Write) - (Unit 6) is the device that receives questions from the program concerning the input of data.

Default Data - (Unit 4) is the file containing program default data.

The SIMAIR program is written in FORTRAN 77. In order to compile this program on your system, it must be able to handle this improved version of FORTRAN.

7. REFERENCES

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3. Duffie, J.A. and W.A. Beckman. Solar Engineering of Thermal Processes. 1980. John Wiley & Sons, Inc. New York.

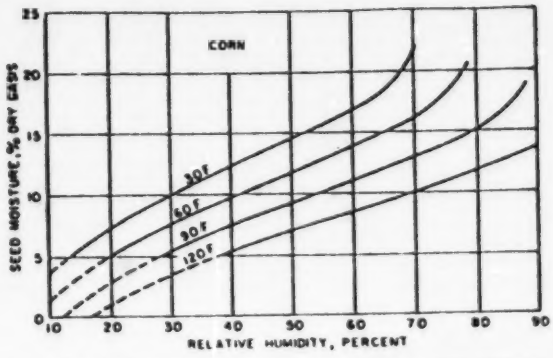
4. Orgill, J.F. and K.G.T. Hollands. 1977. "Correlation Equation for Hourly Diffuse Radiation on a Horizontal Surface". Solar Energy 19(2).

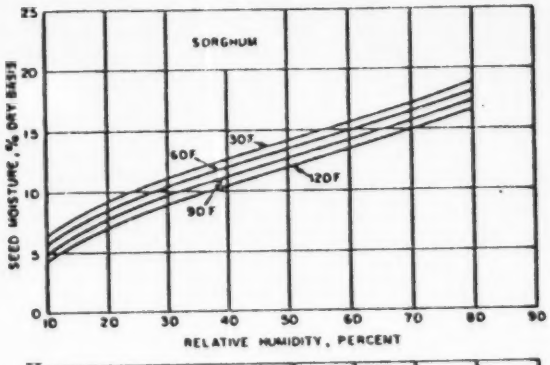
5. Temps, R.C. and K.L. Coulson. 1977. "Solar Radiation Incident upon Slopes of Different Orientation". Solar Energy 19(2).

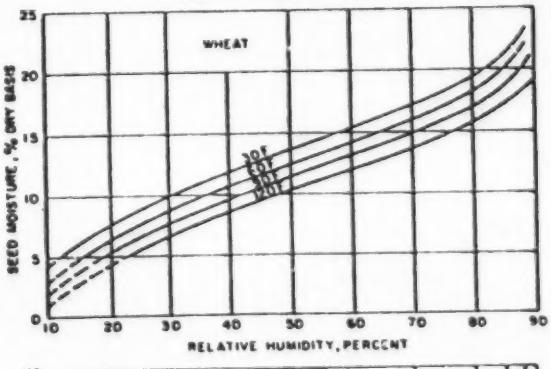
6. Klucher, T.M. 1979. "Evaluation of Models to Predict Insolation on Tilted Surfaces". Solar Energy 23(2).

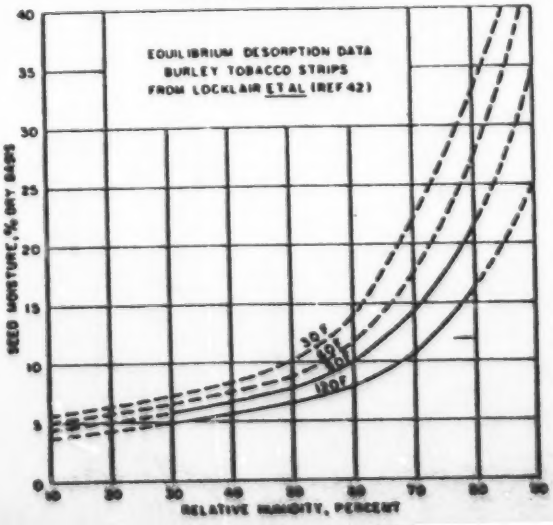
7. Morton, B. and S. Carpenter. 1981. "Use of Air-based Solar Collectors to Preheat Ventilation Air". 1981 Solar Energy Society of Canada Inc. Conference. Montreal.

APPENDIX A: EQUILIBRIUM MOISTURE RELATIONSHIPS FOR CERTAIN CROPS taken from the 1977 Ashrae Handbook of Fundamentals









MAP
Canadian Weather Stations
Resolute
Frobisher Bay
Churchill
Goose Bay
Gander
St Johns
Stephenville
Sept. Iles
Prince George
Edmonton
Saskatoon
Swift Current
Medicine Hat
Brandon
Winnipeg
Kapuskasing
Thunder Bay
Val d'Or
Riviere du Loup
Bagotville
Quebec
Charlo
Chatham
Fredericton
Saint John
Sydney
Charlottetown
Moncton
Truro
Halifax
Yarmouth
Sherbrooke
Montreal
Sault Ste Marie
Sudbury
Ottawa
Muskoka
Kingston
Toronto
London
Vancouver
Victoria
Summerland

City	Province	Latitude (Deg.)	Year	Solar Rad. Derived/Measured
Victoria	B.C.	48.7	1971	D
Prince George	B.C.	53.9	1974	M
Vancouver	B.C.	49.2	1971	M
Summerland	B.C.	49.6	1971	D
Frobisher Bay	N.W.T.	63.8	1975	D
Resolute	N.W.T.	74.7	1971	M
Edmonton	Alta.	53.6	1971	M
Medicine Hat	Alta.	50.0	1971	D
Uranium City	Sask.	59.6	1971	D
Swift Current	Sask.	50.3	1971	D
Saskatoon	Sask.	52.2	1971	D
Churchill	Man.	58.8	1975	D
Brandon	Man.	49.9	1971	D
Winnipeg	Man.	49.9	1971	M
The Pas	Man.	53.8	1971	M
Thunder Bay	Ont.	48.4	1971	D
Sault Ste. Marie	Ont.	46.5	1971	D
Sudbury	Ont.	46.5	1971	D
Kapuskasing	Ont.	49.4	1966	M
Kingston	Ont.	44.2	1971	D
Muskoka	Ont.	45.0	1971	D
Windsor	Ont.	42.3	1971	D
London	Ont.	43.0	1971	D
Toronto	Ont.	43.7	1971	M
Ottawa	Ont.	45.4	1971	M

Montreal	Que.	45.5	1971	M
Sept. Iles	Que.	50.2	1974	M
Quebec	Que.	46.8	1971	D
Sherbrooke	Que.	45.4	1971	D
Riviere du Loop	Que.	47.8	1971	D
Bagotville	Que.	48.3	1971	D
Val D'Or	Que.	48.0	1971	D
Fredericton	N.B.	45.9	1971	M
Charlo	N.B.	48.0	1971	D
Chatham	N.B.	47.0	1971	D
Moncton	N.B.	46.1	1971	D
St. John	N.B.	45.3	1971	D
Charlottetown	P.E.I.	46.3	1971	D
Truro	N.S.	45.4	1971	D
Halifax	N.S.	44.7	1971	M
Sydney	N.S.	46.2	1971	D
Yarmouth	N.S.	43.8	1971	D
St. John's	Nfld.	47.6	1971	M
Gander	Nfld.	49.0	1971	D
Stephenville	Nfld.	48.5	1971	D
Goose	Nfld.	53.3	1971	M

POOR

QUALITY

ORIGINAL